PRODUCT ENGINEERING

NOVEMBER 1950

UNIVERSITY OF MICHIGAN
NOV 14 1950
ENGINEERING LIBRARY

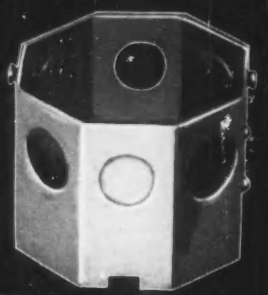
GM's J. O. Almen: "FATIGUE WEAKNESS OF SURFACES"

ALSO IN THIS ISSUE — Centrifugal Castings + Mechanical Buffers + Shakers for Vibration Analysis + Thyratron Timing Circuits + Design Data on Rubber

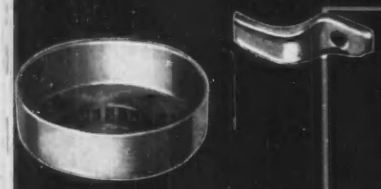
A McGRAW-HILL PUBLICATION

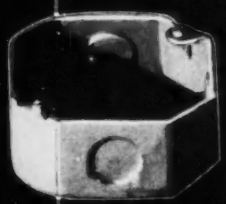
GALVANITE*

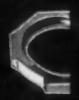
MAKES BETTER PRODUCTS FOR ELECTRICAL MANUFACTURERS



The special tight band of zinc to steel makes Galvanite* the most desirable material where severe fabrication of rust-repelling steel is necessary. The finish stays put—cutting reject losses and adding to sales appeal. It doesn't flake or peel or coat dies—saving production time and costs. For the same reasons it won't fail—bringing everlasting customer satisfaction. For electrical hardware—or anything else that requires zinc coat steel—there's nothing better than Galvanite*.



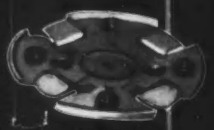




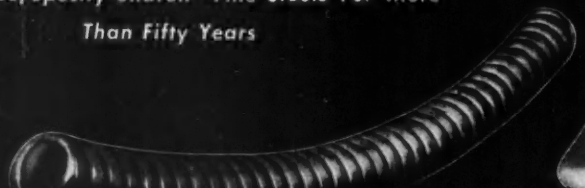
Galvanite Is NOT Shiny

The special patented process employed in the manufacture of Galvanite* gives the steel a grey finish. Do not expect a shiny, tinny appearance. Most products made of Galvanite* are hidden or painted—therefore Sharon makes no attempt to beautify the steel. Only high quality at standard prices are considered in its manufacture.

* T. M.



For Quality Electrical Sheet, Stainless, and Special Alloys, Too, Specify Sharon—Fine Steels For More Than Fifty Years



SHARON STEEL CORPORATION

Sharon, Pennsylvania

PRODUCTS OF SHARON STEEL CORPORATION AND SUBSIDIARIES: THE NILES ROLLING MILL COMPANY, NILES, OHIO, DETROIT TUBE AND STEEL DIVISION, DETROIT, MICHIGAN; BRAINARD STEEL COMPANY, WARREN, OHIO; SHARONSTEEL PRODUCTS COMPANY, DETROIT, MICHIGAN, AND FARRELL, PENNA.; CARPENTERTOWN COAL & COKE CO., MT. PLEASANT, PENNA.; FAIRMONT COKE WORKS, FAIRMONT, W. VA.; MORGANTOWN COKE WORKS, MORGANTOWN, W. VA.; JOANNE COAL COMPANY, RACHEL, W. VA. Hot and Cold Rolled Stainless Strip Steel—Alloy Strip Steel—High Carbon Strip Steel—Galvanite Special Coated Products—Cooperage Hoop—Electrical Steel Sheets—Hot Rolled Annealed and Deoxidized Sheets—Galvanized Sheets—Enameling Grade Steel—Welded Tubing—Galvanized and Fabricated Steel Strip—Steel Strapping, Tools and Accessories.

DISTRICT SALES OFFICES: Chicago, Ill., Cincinnati, O., Cleveland, O., Dayton, O., Detroit, Mich., Indianapolis, Ind., Milwaukee, Wis., New York, N. Y., Philadelphia Penna.

VOLUME 21 · NUMBER 11

PRODUCT ENGINEERING

G. F. NORDENHOLT, *Editor*

Assistant to the Editor, Joseph Kerr • *Managing Editor,* V. H. Laughner • *Associate Editors,* Harold Schot J. D. Cooney • *Assistant Editors,* E. S. Cobb, P. F. Horbatt, P. O'Keefe, Jr., T. M. Warwick, N. J. Van Ness
Consulting Editor, Robert E. Bruckner • *Washington,* George B. Bryant, Jr.
Publisher, Albert Hauptli • *Assistant Publisher,* W. E. Kennedy

Foreign News Bureaus: London • Paris • Berlin • Moscow • Tokyo • Melbourne • Rio de Janeiro • Buenos Aires

NOVEMBER 1950

Timing Machine Operations With Small Thyratrons by Stuart C. Rockafellow 85
Design and application of electronic circuits with time delay characteristics.

Pneumatic vs Hydraulic Systems for Aircraft by H. F. Gerwig and J. H. Famme 91
Performance and design comparisons of systems for the "Convair-Liner".

Product Designs 96
Heavy duty dump truck; Waterproof battery ignition system; Conveyor system for gear finisher.

Engineering Design with Natural and Synthetic Rubbers by Ellwood F. Riesing 106
Properties of fifteen important types of elastic polymers.

Centrifugal Castings in Permanent Molds by Warden F. Wilson 112
Properties and design characteristics of centrifugally cast alloy steels.

Fatigue Weakness of Surfaces by J. O. Almen 117
An analysis and evaluation of the weakness of the surfaces of engineering materials in fatigue. New concepts relative to the strength of materials that reveal shortcomings in current testing procedures and preparation of laboratory fatigue test specimens from which design data are derived.

Telephone Type Relays—Coil Constructions and Timing by V. E. James 141
Speed performance and life characteristics of typical relay coils and stepping switches.

Construction Details of Over-Riding Clutches by A. DeFeo 148
Important design features of several clutches are illustrated.

Applying Shakers for Product Vibration Analysis by Robert C. Lewis 150
Characteristics of electrodynamic shakers and their uses for four types of testing.

Solving the Pythagorean Principle on the Slide Rule by H. W. D'Entremont 155
A simple method for figuring forces, electrical values and linear dimensions.

Selection Factors for Mechanical Buffers by Joseph Brown 156
Design and selection of devices for stopping moving bodies without damage.

Properties of Rectangular Sections by C. A. Kulmann 167
Nomographic chart saves time in computing areas, section moduli, and moment of inertias.

Conversion Chart for Angular and Linear Tolerances by Kenneth F. Stone 169
Eliminates troublesome calculations in dimensioning drawings.

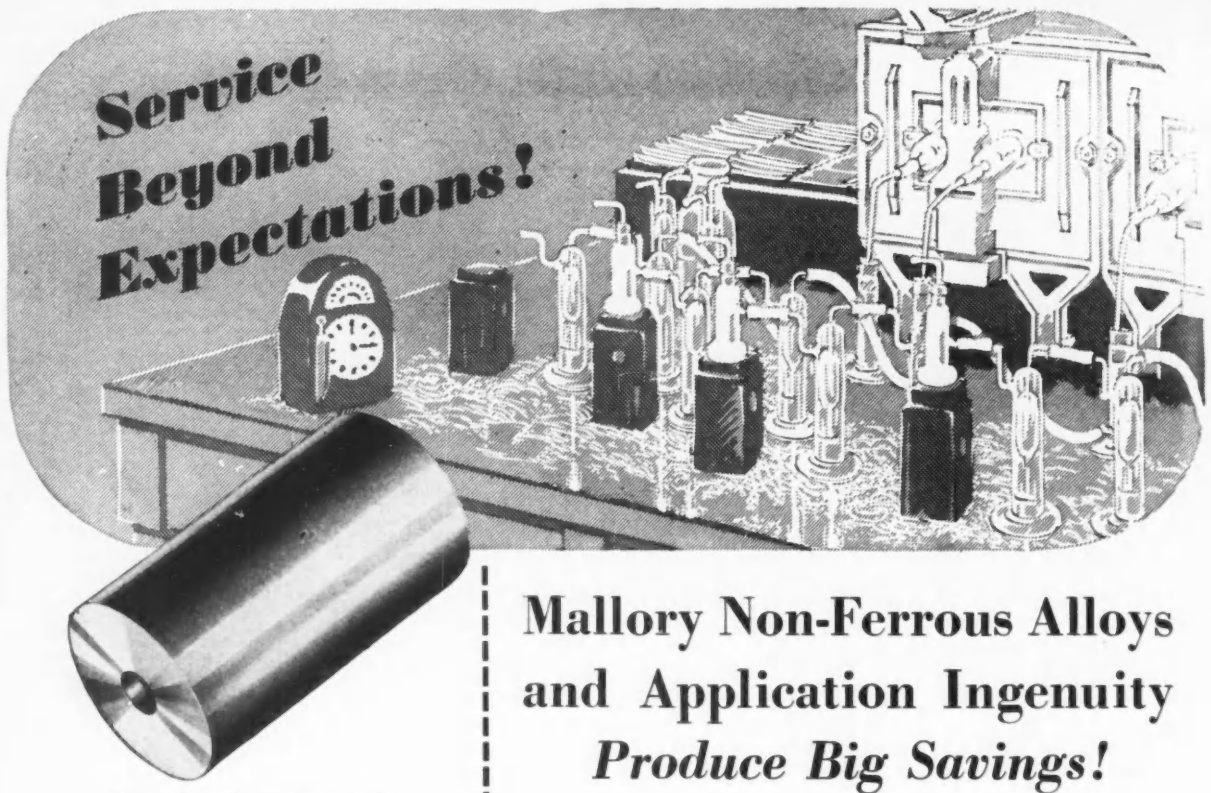
Column Nomograph by C. A. Kulmann 171
For obtaining the stress in median length columns made of pipes or solid sections.

This Month's Cover 9
Editorial 83
Engineering Abstracts 145
Technical News 161
New Parts and Materials 172
New Catalogs and Bulletins . . 208
Our Readers Say 212
Meetings 226
New Books 228

FOR MORE INFORMATION: READER SERVICE CARD PAGE 188

Publication Office: 99-129 N. Broadway, Albany 1, N. Y.
Published Monthly by McGraw-Hill Publishing Company, Inc., James H. McGraw (1860-1948), Founder; Curtis W. McGraw, President; Willard Chevalier, Executive Vice-President; Paul Montgomery, Senior Vice-President. Publications Division: Ralph B. Smith, Editorial Director; Nelson Bond, Vice-President and Director of Advertising; Joseph A. Gerardi, Secretary and Treasurer; J. E. Blackburn, Jr., Vice-President and Director of Circulation.
Correspondence About Subscriptions should be addressed to Director of Circulation, Product Engineering, 330 W. 42nd St., New York 18. Allow at least 10 days for change of address. Indicate position and company connection on all subscription orders.

Subscription Rates: Single copies 75c. United States and possessions, $6 for one year, $9 for two years, $12 for three years; Canada, $7 or one year, $11 for two years, $14 for three years; Latin American countries, $10 for one year, $16 for two years, $20 for three years; all other countries, $15 for one year, $25 for two years, $30 for three years.
Editorial and Executive Offices: 330 W. 42nd St., New York 18. Cable address: "McGraw-Hill," N. Y.
Branch Offices: ATLANTA, R. M. Malutsby, 1311 Rhodes-Haverty Bldg.; BOSTON, M. A. Williamson, Jr., 1427 Statler Bldg.; CHICAGO, Anson E. Meanor, 520 N. Michigan Ave.; CLEVELAND, A. F. Tischer, 1510 Hanna Bldg.; DALLAS, J. H. Allen, First National Bank Bldg.; DETROIT, P. G. Weatherby, 2980 Penobscot Bldg.; LOS ANGELES, C. W. Dysinger, 1111 Wilshire Blvd.; NEW YORK, J. A. McGraw and D. C. McGraw, Jr., 330 W. 42nd St.; PHILADELPHIA, David G. Jones, Architects Bldg.; PITTSBURGH, A. F. Tischer, 738-9 Oliver Bldg.; SAN FRANCISCO, J. W. Otterson, 68 Post St.; WASHINGTON, National Press Bldg.; LONDON, Aldwych House, Aldwych, W. C. 2, Eng. Entered as second-class matter Aug. 21, 1936, at Post Office Albany, N. Y., under act of Mar. 3, 1879. Printed in U. S. A. Member of A.B.P., A.B.C. Copyright 1950 by McGraw-Hill Publishing Co., Inc. All rights reserved.



MALLORY 1000 AND OTHER ALLOYS

Mallory 1000 is a high density metal whose density approaches that of tungsten and platinum. It has great strength and is readily machinable. Its resistance to the corrosive action of many acids, as well as the atmosphere, is superior.

Mallory is a leading producer of unusual alloys and materials from copper, silver, platinum, palladium, tungsten, molybdenum, and other non-ferrous metals and alloys. Recently the line has been broadened to include certified pure copper, aluminum-bronze, nickel and nickel alloys. This places Mallory in a qualified position to make effective and impartial recommendations for materials best suited to your needs. Write today.

Mallory Non-Ferrous Alloys and Application Ingenuity *Produce Big Savings!*

A recent experience again illustrates the ingenuity of Mallory metallurgical engineers . . . and the unique characteristics of one of the many non-ferrous alloys at their command.

Finding the density and physical characteristics of lead unsatisfactory for a unique radium container, an instrument maker was about to turn to gold or platinum. Mallory stepped in . . . demonstrated that readily machinable, high density Mallory 1000 Metal was equal to the job. The entire cost of the Mallory 1000 was less than that of the insurance alone on gold or platinum.

That's service beyond expectations!

And whether your problem is metallurgical or electronic, what Mallory has done for others can be done for you!

In Canada, made and sold by Johnson Matthey & Mallory, Ltd., 110 Industry St., Toronto 15, Ontario

Metallurgical Products and Alloys

P. R. MALLORY & CO. Inc.

MALLORY

P. R. MALLORY & CO., Inc., INDIANAPOLIS 6, INDIANA

SERVING INDUSTRY WITH

Capacitors	Contacts
Controls	Resistors
Rectifiers	Vibrators
Special Switches	Power Supplies

Resistance Welding Materials

Facts and Fancies

DESIGN ENGINEERS are always making assumptions. They must. As one highly successful designer put it, "The job is largely one of intelligent guessing". The emphasis is on the word intelligent. Ironically, designers too often forget that rules and formulas are also based on assumptions.

Empirical formulas and rules of proportioning are based on many assumptions. So are rationally derived formulas. This is soon forgotten and engineers accept the formulas as iron clad relationships. Rules become looked upon as the very laws of nature.

No designer can reach great heights in the creation of new products if he fails to analyze and weigh the facts. He will often find them to be fancies. The diameter of a pulley should be ten to twenty times the thickness of the belt running over it, according to the rules. But a diameter of only six times the belt thickness is sometimes enough. For a given weight per foot a hollow shaft is stronger, according to "rules". But this is only a fancy because it is a fact that under torsional impact a solid shaft may be the stronger. Forgings, according to fancy, have more consistent strength properties than castings, but the reverse is a fact when certain process and inspection controls are applied in the manufacture of the castings.

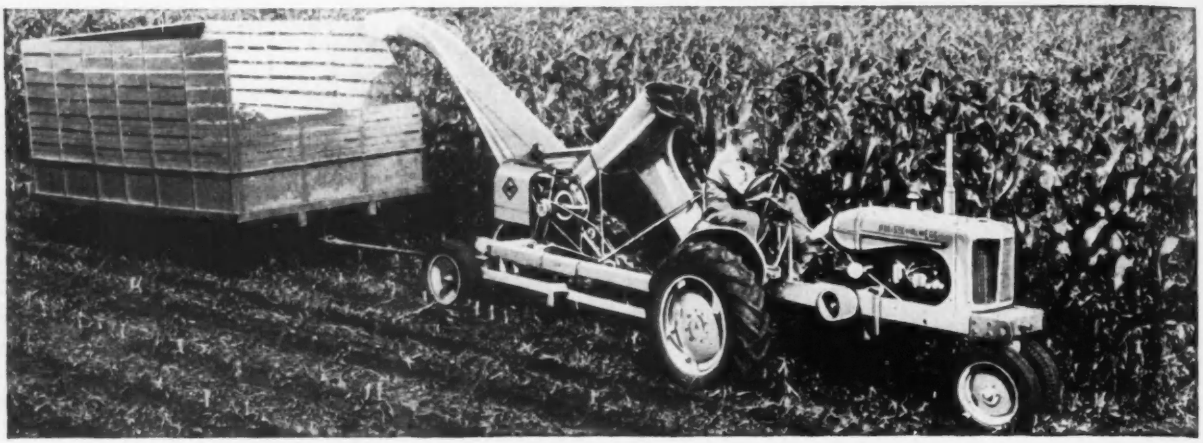
Many more examples can be cited wherein the human mind has the tendency to accept empirical relationships as iron clad fact. Even more prevalent is the habit of not analyzing the basic assumptions on which a rational equation was derived.

In addition to assumptions relating to design calculations, there are also the assumptions relating to costs. It is a fact that aluminum and stainless steels cost several times as much per pound as does sheet steel. But it is only a fancy that an aluminum or stainless structure will cost correspondingly as much more than a steel one. It might cost less. Die castings in either zinc or aluminum cost more per pound than sand castings but usually show considerable cost savings wherever used.

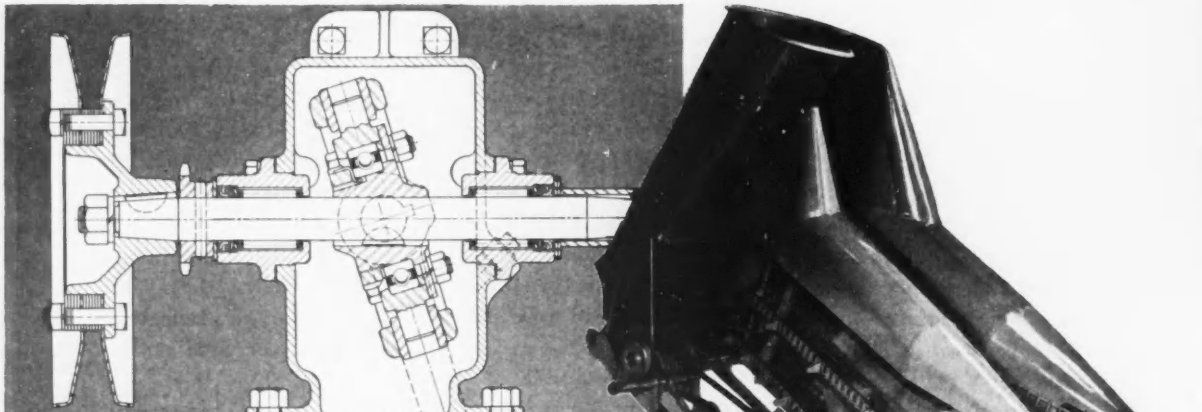
At every step in the design procedure from the original conception to the finished design and specifications, the engineer is faced with facts, assumptions and fancies. Every move the product engineer makes must be studied and weighed carefully in the light of knowledge and experiences. Almost invariably, compromises must be made and preconceived notions discarded.

Notable design developments are not created by minds that blindly accept established rules and formulas as rigid facts. The great creations in engineering were brought about by men who refused to have their aims defeated by conventionalisms. They put their minds to work analyzing every detail, questioning every assumption and developing new design concepts. Such men refused to admit defeat merely because some great minds of the past, working under the then existing limitations of knowledge and experience, had indicated it could not be done. And out of the labors of such pioneers have come, and always will come, the greatest developments. For they recognize that tomorrow the facts of today may just be fancies.

G. F. Rodenhold



Double Action Bearing



TRANSLATING rotary motion into reciprocating motion for the sickle drive on the row-crop attachment for the Allis-Chalmers Forage Harvester is not a job for a bearing that can't put up with variety in its life—a daily round of dizzy motions and alternating loads.

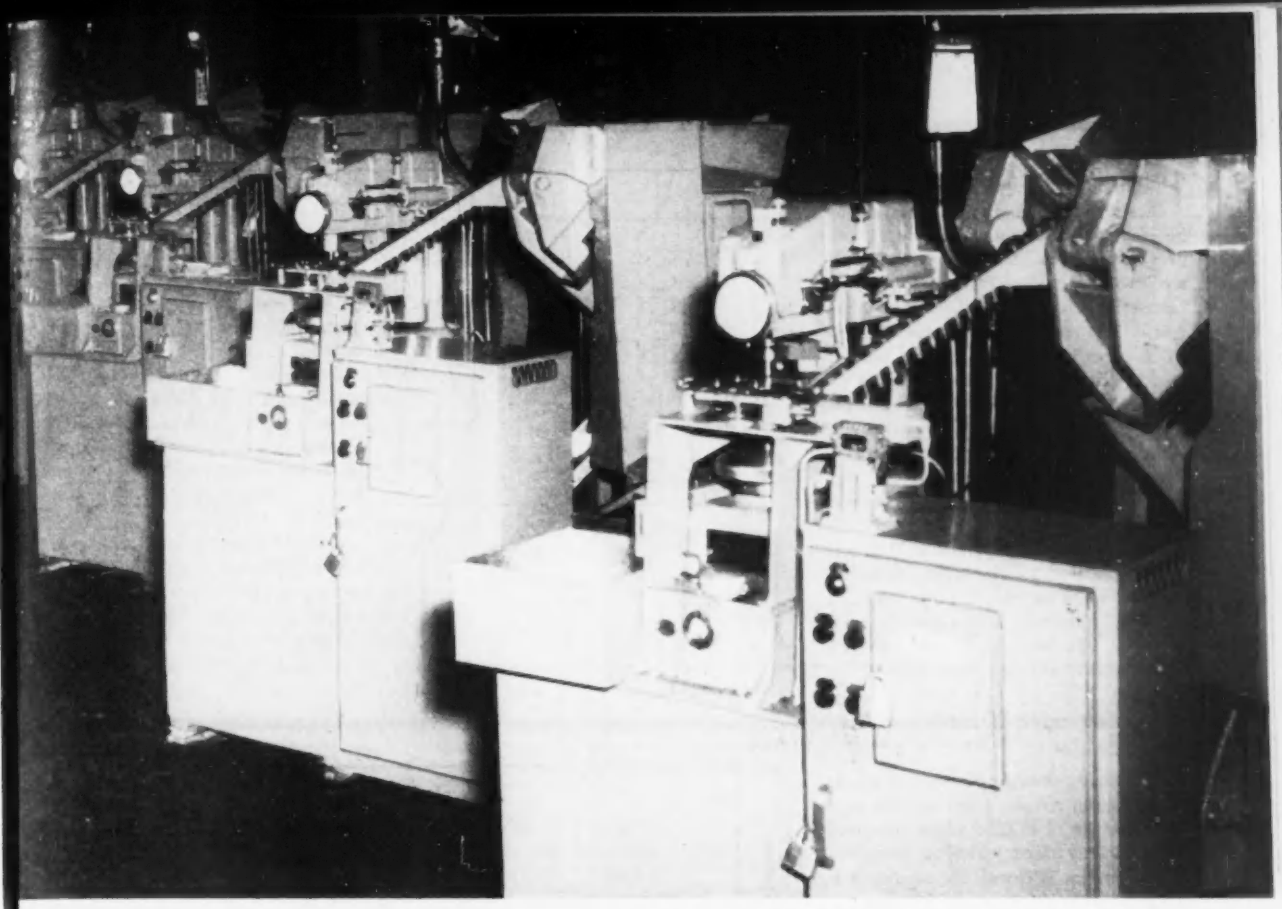
One compact New Department single row ball bearing does the job because taking thrust in both directions as well as radial load is fundamental to its design. Without need for adjustments, this bearing uniquely combines the required internal rigidity with utmost freedom of rotation, and here it is fully enclosed against dirt—needs minimum attention for lubrication.

Nothing Rolls Like a Ball

NEW DEPARTURE BALL BEARINGS

NEW DEPARTURE · DIVISION OF GENERAL MOTORS · BRISTOL, CONNECTICUT

117-C



THYRATRON TIMING DEVICE controls the performance of these automatic production machines. The process involves five operations on each piece and all operations must be precisely timed.

Timing Machine Operations With

SMALL THYRATRONS

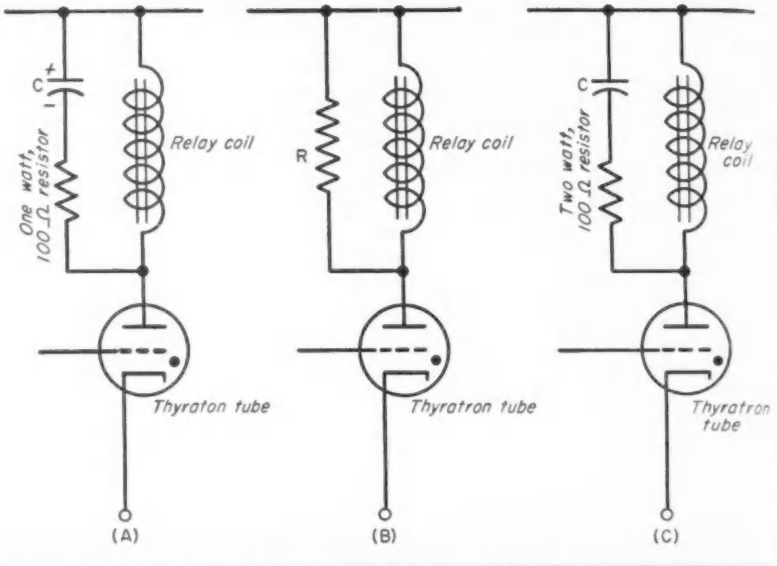
STUART C. ROCKAFELLOW
Robotron Corporation

CIRCUITS BUILT AROUND THYRATRON TUBES are often the most economical means of introducing time delay into the control systems of automatic or semi-automatic machines. The basic circuit consists of a thyratron having a condenser connected to its grid. The tube cannot trigger (conduct) as long as the charge on the condenser is of sufficient value. But when the charge has been reduced through leakage, the thyratron fires. The condenser leakage is responsible for the delayed action of the thyratron circuit.

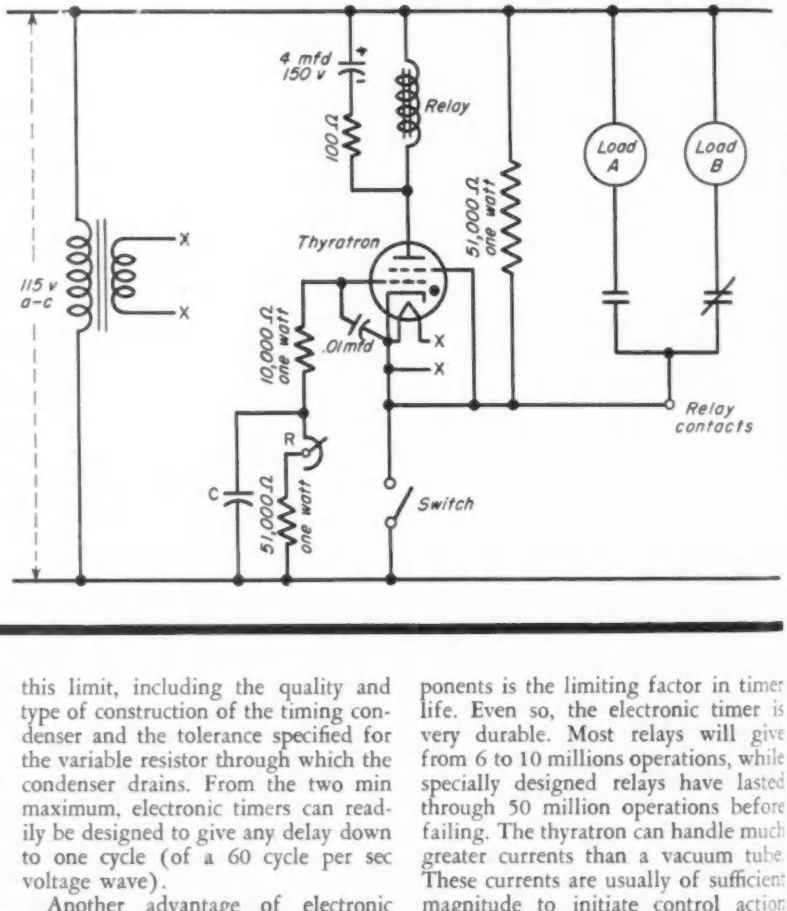
Resistance welders, plastic presses, punch presses and automatic test equipment are typical machine applications requiring time delays in their control. Timed functions can be controlled by several other means. One uses a synchronous motor to drive a cam that closes a contact at some point in its rotation. Another is a dash pot arrangement with a variable orifice to control the escape of air until the piston hits bottom. The piston actuates electrical contacts that either open or close a circuit. The grid control characteristics of vacuum tubes are also used for timing. When the negative side of a charged condenser is placed on the grid of a vacuum tube and the condenser allowed to drain off through a variable resistor, some time will be required for the negative bias to reach a low enough value for the tube to conduct and to pull in a relay.

Each method has its limitations. The cam type timer is ordinarily hard to adjust to short times such as fractions of a second but is readily adaptable to long timing ranges. The pneumatic controls are easy to adjust but require air lines leading to the units. The vacuum tube type of timer has easily

1 Although thyratrons can be made to control a load directly, relays must be employed for currents above 25 amp or where complicated control operations are involved. In Fig. 1(B) are shown the connections for small clapper type a-c relays. The resistor has a value of about 2,000 ohms and 5 watts, and is used to feed the back emf to the coil during the half cycle when current is cut-off. The smaller relays handle loads up to 10 amp easily. Thyratrons can operate large relays directly if added capacity is connected. In Fig. 1(C) are shown the connections required when a 25 amp relay is to be driven by a thyratron. This type of relay will draw the least amount of current when a four mfd capacitor (paper) is shunted across the 115 volt a-c coil. The resistor R prevents high peak currents through the tube.



2 Schematic drawing showing a simple thyratron circuit which may be used to either delay or to hold a load energized for a predetermined period of time. Before the switch is closed, the condenser is charged negatively at the grid end by using the grid as an anode. Electrons pass either through the 51,000 ohm resistor or through the load from the upper line, then from cathode to grid to C, the charge condenser. As the switch is closed, the cathode is shorted to the bottom line and the condenser drains through variable resistor R to a low enough value for the tube to conduct. At this time the relay operates. If load B is connected, it will become energized when the switch is closed and remain energized until the time delay is over. If load A is connected, it will become energized after the time delay. Thyratrons can operate some loads directly. The coil of a magnetic starter could be substituted for the relay coil.



adjustable timing, but is usually subject to line voltage variations that affect its time setting.

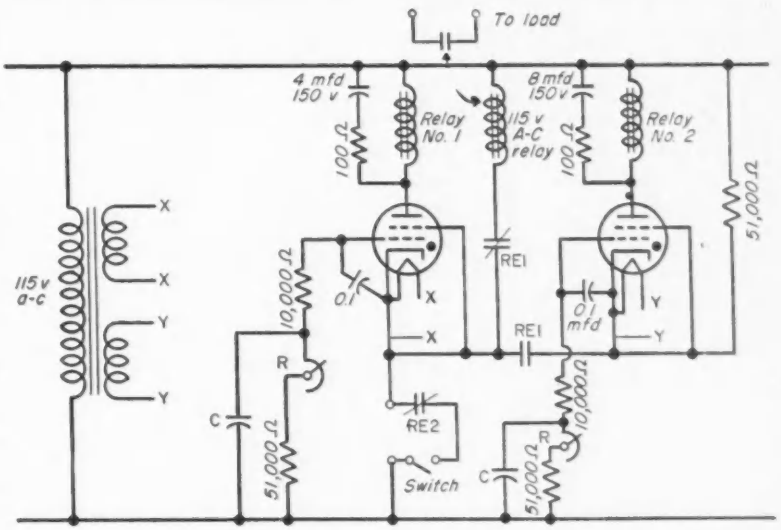
The thyratron is an extremely rugged tube and, when used in a correctly designed circuit, will hold its timing to small variations even with fairly large supply voltage changes. One limiting factor of both vacuum and thyratron tube timers is the length of timing. Two min is usually recommended as the maximum delay to be obtained with either method. It is possible to make these timers give delays of much longer periods, but not with good repetitive accuracy. Many factors govern this limit, including the quality and type of construction of the timing condenser and the tolerance specified for the variable resistor through which the condenser drains. From the two min maximum, electronic timers can readily be designed to give any delay down to one cycle (of a 60 cycle per sec voltage wave).

Another advantage of electronic timers—both vacuum and thyratron tube—is long service life. Electron tubes will last for many millions of operations. Ordinarily, the mechanical endurance of the pilot relays used in conjunction with the electronic components is the limiting factor in timer life. Even so, the electronic timer is very durable. Most relays will give from 6 to 10 millions operations, while specially designed relays have lasted through 50 million operations before failing. The thyratron can handle much greater currents than a vacuum tube. These currents are usually of sufficient magnitude to initiate control action directly without the use of pilot relays.

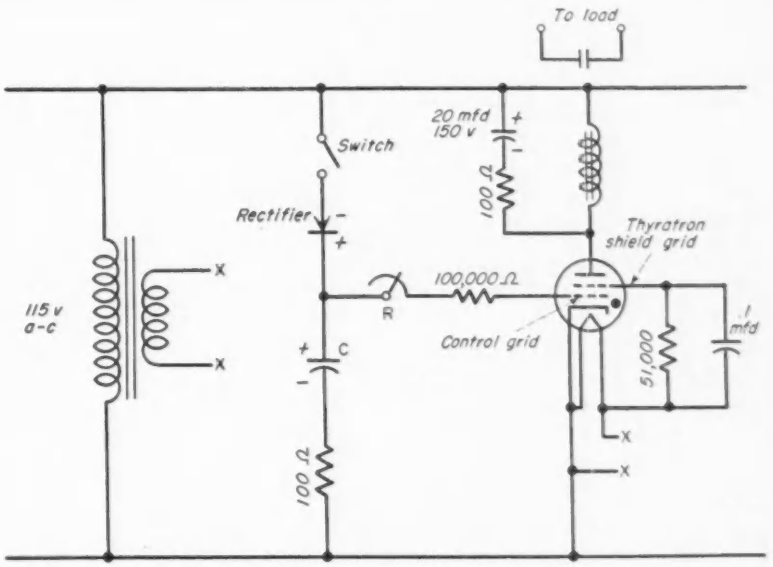
Small Thyratrons

Thyratrons are very similar in appearance to vacuum tubes. Effectively, the internal structures are the same

3 The basic circuit of Fig. 2 can be modified to become an automatic "on-off" timer. One use might be in the application of voltage pulses for insulation breakdown tests. Before the switch is thrown, the timing condensers in the control grids of both thyratrons are charged by conduction, as explained previously. When the switch is closed, the "on" tube triggers after a set time delay. This energizes relay No. 1, cuts out the a-c relay and puts plate voltage on the "off" tube. After another set delay, the "off" tube fires, energizing relay No. 2, cutting off plate voltage from both tubes. Relay No. 1 opens immediately. But the larger (8 mfd) condenser across the coil of relay No. 2 holds this relay closed for an interval sufficient for the charge on the grid condenser of the "on" tube to build up before the cycle begins again.



4 In some applications, a relay must remain closed for a certain length of time after the switch is opened. This applies to some punch press operations, where each piece of ejected stock trips a momentary contact switch. This switch starts the descent of the press ram on the succeeding piece of stock. The momentary closing of the switch causes the thyratron to fire, while a time-delay circuit keeps the tube firing until the stamping operation is completed. Phasing of the filament transformer is important as the shield grid must be made negative when the plate is positive. This keeps the tube blocked. As the switch is closed, a large positive charge is placed on the control grid, overcoming the small negative bias on the shield grid, and the tube fires pulling in the relay. Even if the switch is opened, some time is required for the positive charge to drain until the negative shield again blocks the tube.



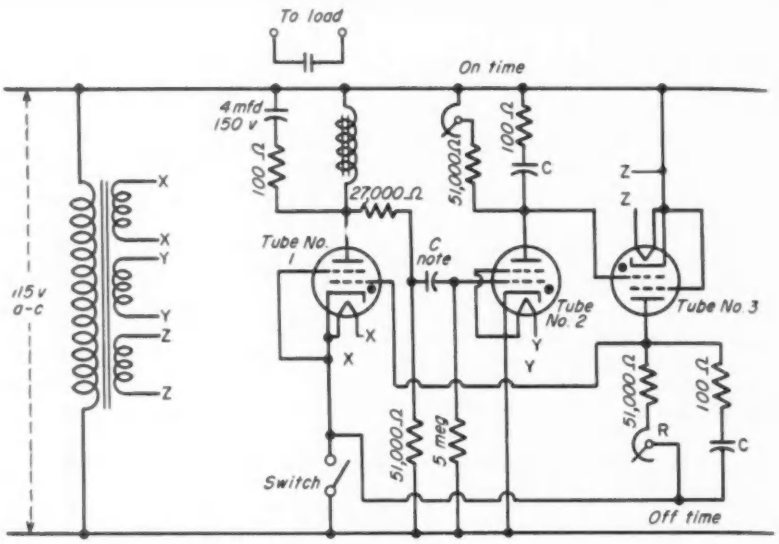
Each has a cathode, one or more grids and an anode. However, while the vacuum tube is completely evacuated, the thyratron is not. During manufacture, the thyratron is evacuated. Then a small amount of gas is introduced into the envelope before sealing. This gas makes the difference between the vacuum and thyratron tubes. In the vacuum tube the grid can control the amount of electron flow between cathode and plate, while in the thyratron the grid can control only the start of the electron flow. In the vacuum tube the bias, or negative voltage on the grid, will control the amount of current the tube passes. With a high negative bias the vacuum tube will pass low current and with a lower negative bias the cathode to anode current will increase. With the thyratron, there will be no electron flow until the grid is at a very low negative value and once the electron flow starts from cathode to anode, the grid has no more control. In usual cases, the only way to stop the thyratron from continued conduction is to reduce the anode supply voltage to zero.

The plate current of a vacuum tube responds to changes in grid voltage within millionths of a second. The thyratron is slower in its control, but passes considerably more current. However, the thyratron is still much faster in its control action than necessary ordinarily for industrial purposes.

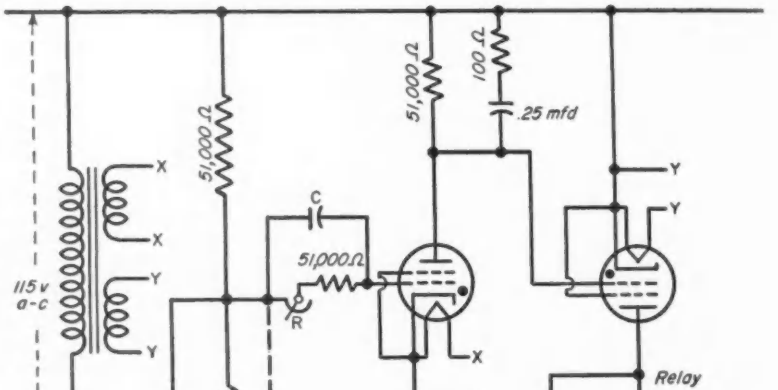
In the thyratron, as in the vacuum tube, the cathode is heated for electron emission. At the start of conduction, plate current is composed entirely of the electrons thus emitted.

In the gaseous atmosphere inside the thyratron, the electrons do not have a free path from cathode to plate. The gas atoms are in the way. The plate potential will determine the speed or velocity that the electrons will

5 This "on-off" circuit needs only one relay. Condenser C-Note in the grid of tube No. 2 is charged negatively at the grid end by grid-cathode conduction from the upper side of the line through the relay. The small current drawn does not energize the relay. When the switch is closed, the relay closes and the charging source for C-Note is effectively shorted to the bottom side of the line. Tube No. 2 becomes blocked, allowing the "ON" time condenser to drain to a low enough value so that tube No. 3 conducts. The "OFF" time condenser immediately places a negative potential on the grid of tube No. 1, blocking that tube and dropping out the relay. As soon as the relay drops out, tube No. 2 again conducts blocking tube No. 3. The "OFF" time condenser now drains to a low enough value so tube No. 1 fires to pull in the relay again.



6 Two thyratrons and one relay make up this timer. The timing condenser is charged by grid-cathode conduction through the first tube. Because there is a small arc drop between grid and cathode, the grid is slightly positive and cathode-anode conduction takes place. This charges the condenser in the anode circuit to a negative potential at the anode side and blocks the second tube. When the switch is closed, the charging potential for the timing condenser is removed, the negative charge is placed on the grid and the first tube is blocked. With the first tube blocked, the negative voltage on the grid of the second tube is removed and the tube fires, pulling in the relay. When the charge on condenser C has drained off, the first tube conducts, placing a negative voltage on the grid of the second tube and the relay falls out. Momentary-contact starting can be provided.



be attracted to the anode. As the potential increases, the speed increases. If this potential is great enough, an electron attracted to the plate will collide with a gas atom with sufficient energy to knock off one of the electrons in the gas atom. Thus, two electrons are free to pass to the anode. This leaves the gas atom with a positive charge and the process is called ionization. The positive ion of gas is attracted back to the cathode and bombards it with enough energy to liberate several more electrons. The one original electron which was liberated by thermal emission may, by ionization and cathode bombardment start several electrons toward the anode. The time necessary to start complete current flow is called ionization time and is usually in the order of microseconds.

The velocity of the electrons is governed by both plate potential and distance. If another element, the grid, is inserted between the cathode and plate, and a negative charge placed on it, the grid will have a pronounced effect on the electron flow to the plate. With a small negative charge on the grid, the electrons cannot attain sufficient velocity to cause ionization. However, once conduction starts, an ionic sheath gathers around the negative grid rendering it useless for control.

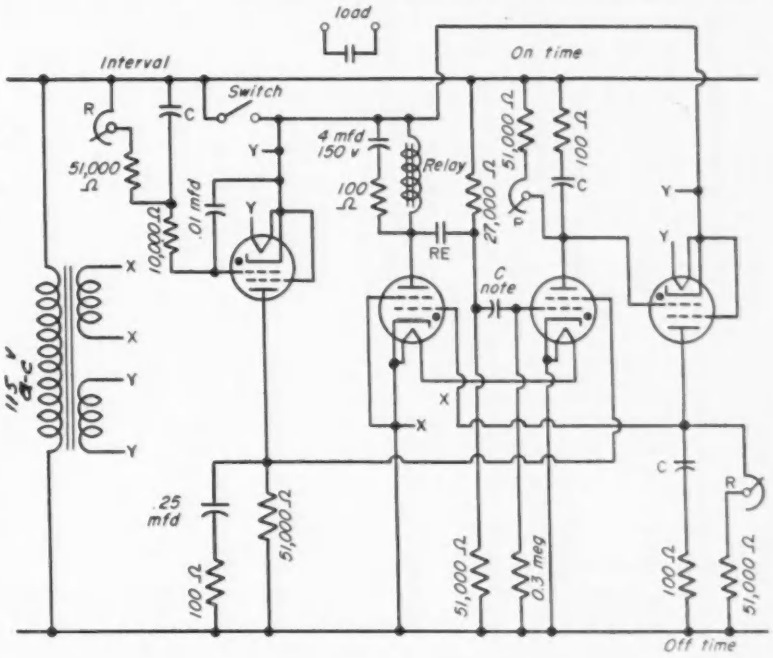
When the anode potential is reduced to zero, the ions or positively charged atoms, begin to regain their original state. Each has to pick up a free electron. The time taken to get back to normal is called "de-ionization time." If the plate voltage is turned on again before complete de-ionization takes place, the tube immediately triggers again. De-ionization time is usually less than one millisecond so the tubes may be used with alternating currents of frequencies up to nearly

1,000 cycles without mis-firing. The use of 60 cycle current insures a 10 to one safety factor in regards to de-ionization.

In general, the thyratron is used as a switch to control passage of current. The grid serves only to delay the start of conduction. Once the tube fires, the grid loses control and conduction can be stopped only by reducing plate voltage to zero. It is for this reason that alternating voltages are ordinarily used in the plate circuit. On the negative half of each cycle, plate current is stopped and the grid resumes control.

Three types of small thyratrons are in general use. The 2D21 is a miniature type; the 2050 has a glass envelope about the size of the ordinary radio receiving tube; while the 502A has a metal case. The characteristics of each are about the same when the shield grid is tied to the cathode. Any one of the three types will safely pass 100 ma continuously, which is sufficient to pull in a fairly large relay. The plate or anode potential has a great effect on the grid control voltage required. If, for instance, the anode potential is 300 volts, the negative grid potential needed to prevent triggering will be more than if anode potential were only 150 volts. With a 115 volt a-c supply the peak plate voltage rises to about 160 volts. In the small type thyratrons, a negative grid voltage of two volts would be sufficient to prevent firing for this magnitude of plate voltage. This latter statement assumes that the screen grid is tied to the cathode. The shield grid may also be employed as a control element. This is not ordinarily done in designing timers. If it were desired to block the tube after a certain number of control sequences or operations, the shield grid could be used to accomplish this.

Because the grids of thyratrons are mechanically strong, they are often made to serve as a combination of both control grid and anode. They are rated to pass 10 ma and in many cases are used to carry the current for charging the timing condensers. Care must be taken in the circuit design to limit the current to the rated amount.

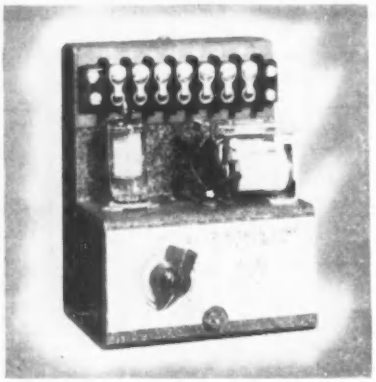


7 Some machines involve operations that must be started and stopped a number of times, then stopped altogether until the next sequence is begun. Such can be accomplished by means of a thyratron timing circuit. For example, it is possible to design an electronically controlled indexing arrangement. A rotary table could be shifted to various stations, with a definite dwell period at each station. After a complete rotation, the table would stop, ready for the next indexing cycle. This requires a combination of a basic timing circuit, such as shown in Fig. 2, with an "on-off" circuit, similar to that of Fig. 3. The basic timer is set for a longer time delay and opens a normally closed relay in the load circuit to stop the sequence of "on-off" operations that are controlled by the other timing section. This set-up requires as many as four relays. With the circuit shown above, only one relay is used to accomplish the same end. Basically, this is the circuit of Fig. 5 with an extra tube to control the duration of the "on-off" sequence. Condenser C-Note must be at least twice as large as the "ON" time condenser. Each of the two filament windings should be rated at least at 1.2 amp as each winding supplies two tubes. The timing for the interval stage can be computed from the tables provided to yield the required number of operations. The "interval stage" in this particular case means the time required for one complete sequence of on-off operations. The duration of the interval stage may be as long as two minutes, providing that thyratron tubes alone are utilized in the timer. Of course, longer intervals can be accomplished by the substitution of a clock-type timer for the thyratron interval timer. Even with the clock timer, thyratrons can be utilized for the individual operations. Combinations of clocks and thyratrons are extremely flexible.

Timer Design

A perfect condenser, once charged, will retain its charge indefinitely. Unfortunately, all condensers have some leakage, which allows the voltage to drain off eventually. When a resistor is placed across the terminals of a charged condenser, the time needed for the charge to drain off is determined by the resistance. If a variable resistance is placed across the terminals of the charged condenser, the time of discharge will be dependent upon the setting of this resistance.

The quality of the condenser and the variable resistor should be high, as these are the basic parts of the timing circuit. Oil filled bathtub-type condensers, non-inductively wound, are generally advised for industrial timing controls. However, many open-type paper condensers have given good results. Electrolytic condensers give poor results on short time periods and are permissible only for long time delays when the results are not critical. Potentiometers are generally used as the variable resistors because of their availability. Due to the low heat dissipation involved, one or two watt sizes are adequate. The accuracy of taper is the most important requirement for the potentiometer or variable resistor. Usually, some means of calibration is necessary and if the taper (variation of resistance with dial setting) of the potentiometer is off, so will be the calibration marking. A straight, linear taper is desirable. In calibrating, the flats of the ends of the potentiometers must be taken into consideration. In the circuits shown, calibration is linear with a straight taper



SMALL TIME DELAY UNIT, employing circuit described in Fig. 2. Indicator dial is calibrated in seconds of time, corresponding to various potentiometer settings. Any number of units may be connected in series for sequence variations.

potentiometer. If the timing is to vary between one sec and 100 sec, then roughly half way around will represent a 50 sec delay.

COMPONENT RATINGS. The product of R (resistance) times C (capacitance) is an indication of the length of time delay incorporated in any thyratron circuit. Thus, to effect a particular time delay, the designer can use any combination of R and C values that give the same product. For instance, when charged through a tube from a 115 volt a-c source, a 0.2 mfd capacitor across a one megohm resistor will delay the triggering of a thyratron for approximately one second. The combination of 0.1 mfd across two megohms or 0.4 mfd across ½ megohms would provide the same time delay. However, on the basis of past performance, certain definite combinations of capacitance and resistance can be recommended for fast repetitive operations. These are typical.

1 megohm	+ 0.2 mfd	=	1 sec
1 megohm	+ 0.5 mfd	=	2½ sec
5 megohms	+ 0.5 mfd	=	12 sec
5 megohms	+ 2.0 mfd	=	50 sec
10 megohms	+ 2.0 mfd	=	100 sec

If the above resistors are variable, the timing may be adjusted from zero to the maximum times shown by varying the resistance. Ordinarily, however, a limiting resistor is placed in series with the variable resistance. This limits the current drawn through the charging device (usually the tube) and also assigns a definite minimum time delay to the circuit. The limiter may be 50,000 ohms in all above cases. This magnitude of resistance establishes the minimum time delay as three cycles (1/1200 sec) with a 0.2 mfd condenser. As the capacity value increases so does the length of the minimum delay.

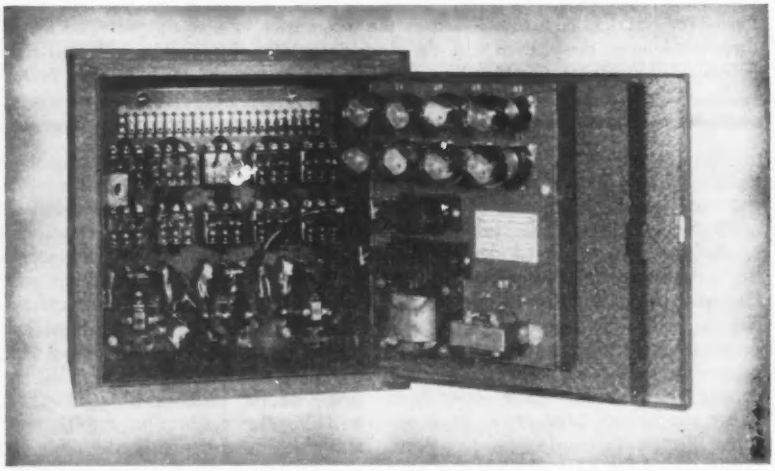
Electronic timing circuits are often designed to fit particular applications. Because of the timing limitations of electronic units, clock or cam type timers are often used in conjunction with electronic timers to provide a series of variable delays. The clock type timer may be used for periods up to several hours, with the electronic timers performing a series of shorter operations during that interval. Several electronic timers may be connected in series to give variable delays for operational sequences. Or, two or more electronic timers could be started at the same time, one to time out before the others. Several timers may be started simultaneously to control a series of operations, the sequence of which depends upon the delay settings of various timers. Many combinations are possible by correct applications of basic circuits.

RELAY REQUIREMENTS. Most timer circuits use a relay as the end means of control. Some of the circuits to be described here will have a small relay to energize a medium sized contactor. From a life and economic standpoint it is usually desirable to design tubes into a circuit wherever possible. If a relay can be replaced by the use of an additional tube then a possible mechanical failure is eliminated. These small thyratrons cost from $1.80 to $2.00 each, while pilot relays may cost from $5.00 to $10.00 each.

Table I—Typical RC Combinations

Time	Condenser	Potentiometer
1 sec	.2 mfd.	1 megohm
	.1 mfd.	2 megohm
	.5 mfd.	400,000 ohm
	1.0 mfd.	200,000 ohm
5 sec	1.0 mfd.	1 megohm
	.5 mfd.	2 megohm
	.2 mfd.	5 megohm
	.1 mfd.	10 megohm
25 sec	5.0 mfd.	1 megohm
	2.5 mfd.	2 megohm
	1.0 mfd.	5 megohm
	.5 mfd.	10 megohm
50 sec	5.0 mfd.	2 megohm
	2.0 mfd.	5 megohm
	1.0 mfd.	10 megohm
100 sec	4.0 mfd.	5 megohm
	2.0 mfd.	10 megohm

Wherever possible, the circuits discussed have used the 115 volt a-c line as a plate supply source with separate filament supply transformers to each tube. Filaments of all three small type thyratrons are rated at 0.6 amp on 6.3 volts. Several types of relays may be used in thyratron control circuits. With the correct components, either a-c or d-c relays will prove satisfactory. In Fig. 1A are shown the connections for incorporation of a d-c relay. The relay may have from 2,500 to 6,500 ohms resistance. Because the thyratron will be working only on every other half cycle, a fairly large condenser must be used to hold the relay closed for the period when no energy is supplied. Electrolytic condensers are usually recommended for this purpose. Polarity markings must be observed as shown. The small one watt resistor (about 100 ohms) is used to prevent high peak currents through the tube during initial conduction. The caption of Fig. 1 describes the elements of other relay circuits.



INTERIOR OF THYRATRON TIMER used to control sequence of various operations in hardness testing. Timer is a combination of several of the circuits described in this article. A clock timer could be incorporated in this controller for long delays.

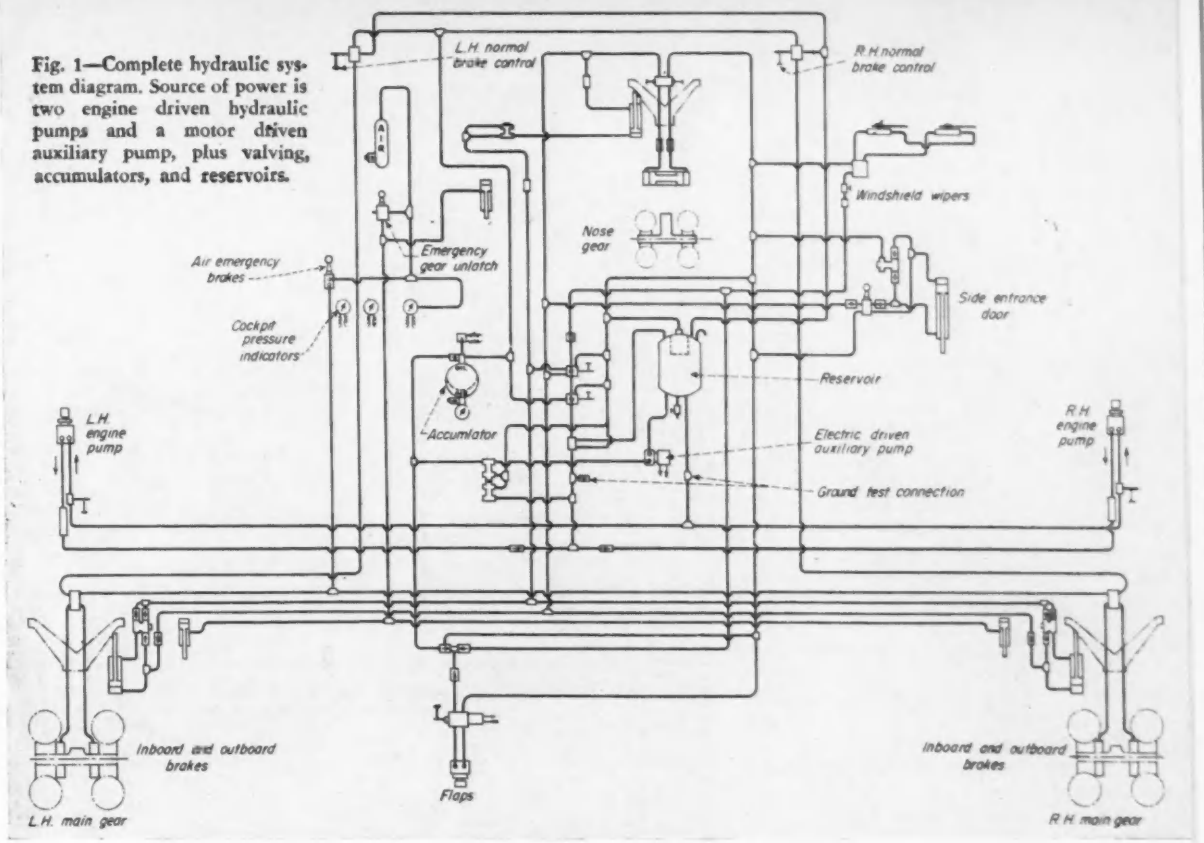


Fig. 1—Complete hydraulic system diagram. Source of power is two engine driven hydraulic pumps and a motor driven auxiliary pump, plus valving, accumulators, and reservoirs.

Pneumatic versus Hydraulic Systems for Aircraft

Actual comparisons of performance characteristics and design of pneumatic vs. hydraulic systems for a given piece of equipment are not often available . . . but here is a detailed case-history study on the "Convair Liner." The comparison is worth study by designers.

H. F. GERWIG
Pneumatics Design Engineer
Convair

J. H. FAMME
Project Engineer

WHEN HIGH PRESSURE PNEUMATIC SYSTEMS have been adopted for component actuation during the initial design stage of any aircraft, less weight, faster action, and greater overall safety generally have been the result. These advantages also can be obtained by the redesign of existing systems, although the problem becomes one of compromise because of the limitations involved: sizes of components may already be at a maximum, the mechanisms already limited as to travel and direction, and time and sequence foregone conclusions. These same problems exist when redesigning any type of pneumatically or hydraulically-operated equipment.

The questions that then come to mind—"What are the important items that must be considered in such redesign?" and "Is the gain in performance worth the effort?"—can best be answered by an actual comparison between hydraulic and pneumatic systems, in this case as applied to a "Convair" Model 240 airplane, which is a two-engined, low-wing monoplane, powered with R-2800 reciprocating engines. The gross weight is approximately 40,000 lb and the airplane is designed to carry 40 passengers. Cabin pressurization is available for operation up to 20,000 ft altitude.

Power System

HYDRAULIC. A 3,000 psi hydraulic system, Fig. 1, provides the power to operate the following components: nose and main landing gear; flaps; brakes; nose wheel steering; side or rear entrance doors; windshield wipers; and cabin pressurization system components. Line sizes and lengths of tubing are not shown in the diagram but are noted and compared with the pneumatic system in the

table. The source of hydraulic power is the two engine-driven hydraulic pumps, one on each engine, plus an electric driven auxiliary pump. The two engine-driven pumps have a total capacity of 16 gpm at 3,000 psi during take-off, and the electric pump has a capacity of 0.75 gpm at 3,000 psi.

Fluid is taken from a 5-gallon reservoir and pumped to the regulating system. The unloader valve permits the system pressure to reach 3,100 psi, and then diverts the full flow to the reservoir return. When pressure drops to 2700 psi, the unloader again diverts flow to the pressure system. To relieve pump load that would occur due to continuous system internal leakage during flight, a by-pass valve is employed to divert flow to the reservoir. This valve also traps the fluid in the accumulator, saving it for brake operation in case of power system failure. Main and auxiliary system relief valves provide protection against excessive pressures; the former is set to crack at 3,250 psi and the latter at 2,900 psi. A 120 cu. in. accumulator with a 1,000 psi initial air pressure charge, is employed to store energy for brake operation, steering, flaps, entrance door and the initial gear retraction.

PNEUMATIC. The pneumatic system for the same airplane is shown in Figs 2 and 3. Sources of pneumatic power are the two engine-driven compressors. They are a three-stage, piston type with a capacity of 5 cfm of free air at 1,500 psi outlet pressure. (A 3,000 psi outlet pressure compressor is currently under development.) If the inlet air can be supplied from the compressor section of jet or turbo-prop engines, supercharging of the compressor results in a reduction of charging time and weight. With the reciprocating engine-driven compressors, air is drawn in from the atmosphere, compressed and delivered to the pressure regulating system. When pressure reaches 1,500 psi the unloader valve diverts flow to the atmosphere and the compressor runs free.

When the system storage pressure drops to 1,350 psi the unloader valve diverts flow to the pressure system. A relief valve is installed to prevent excessive pressures. Each engine compressor supplies flow first to its respective brake system reservoir, then to the pressure supply manifolding line, and then to the normal system reservoir. This is done by means of by-pass valves and insures that the primary requirement, namely brakes, receives preference. The brake system by-pass valve has free flow into the reservoir and provides interconnection with the rest of the system when the storage pressure is above 800 psi. The normal system by-pass valve allows flow into the reservoir when supply pressure is above 850 psi and prevents back flow. The manifolding pressure supply line by-pass valve allows flow only when the supply pressure is above 800 psi, but allows free flow in reverse.

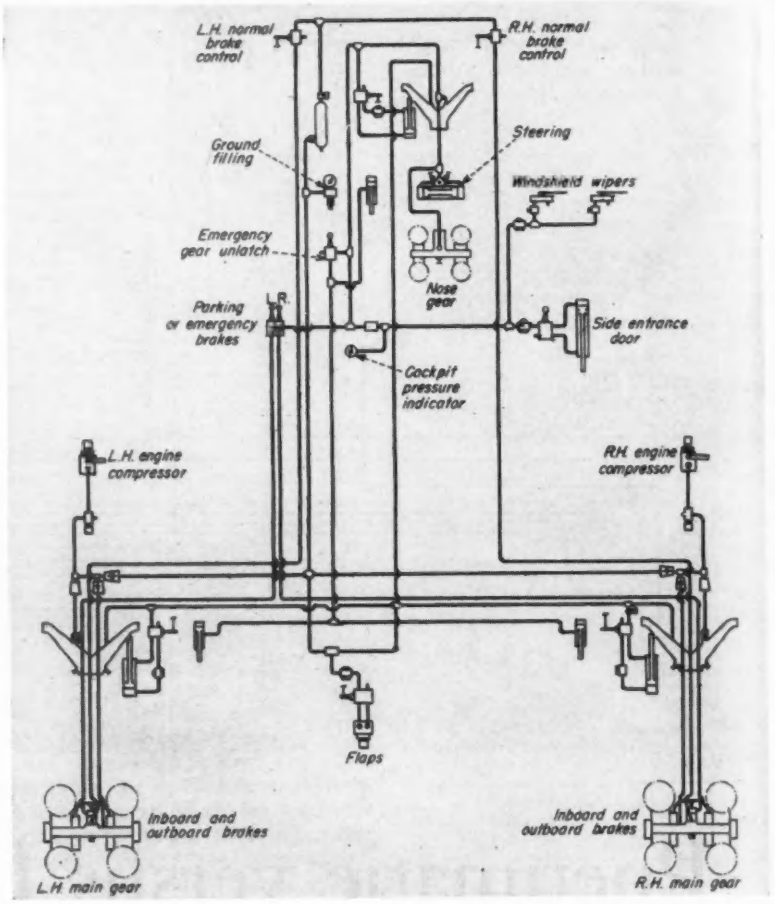


Fig. 2—Complete pneumatic system diagram. Engine-driven compressors of the 3-stage piston type supply air at 1,500 psi. Weight of the pneumatic power system is 49.6 lb as compared to 80.2 lb for the hydraulic system.

Landing Gear

HYDRAULIC. Both the main and nose gears are controlled by one manually operated selector valve. To retract the gears, pressure is applied to the retraction side of the cylinders. The initial motion of the cylinder unlatches the down lock and the gear then retracts. To extend the gear, the up latch is first released by cable control. The gear falls free and locks down with aerodynamic loads alone, so the only emergency equipment necessary is a pneumatic unlatching cylinder that functions in case of cable failure.

PNEUMATIC. A cable operated manual selector valve is provided at each gear to achieve the minimum retraction time with small lines and eliminate the long lines required by the hydraulic system. Return ports from the selector valves are connected to the atmosphere, thus effecting a weight savings in return lines. A viscous damper is built in the actuating cylinder to prevent excessive acceleration and deceleration rates. The latching and unlatching of the gear is identical with that of the hydraulic system.

Nose Wheel Steering

HYDRAULIC. With the use of corotating nose wheels, shimmy-damping is not a problem and requires no viscous damper; therefore it was possible to design a simple steering cylinder with a rack and gear, which would have been prohibitive in a steer and damper unit. The position of the nose wheel is determined by the position of the steering wheel in the cockpit. Motion of this steering wheel actuates the control valve through cables attached to the steering cylinder piston.

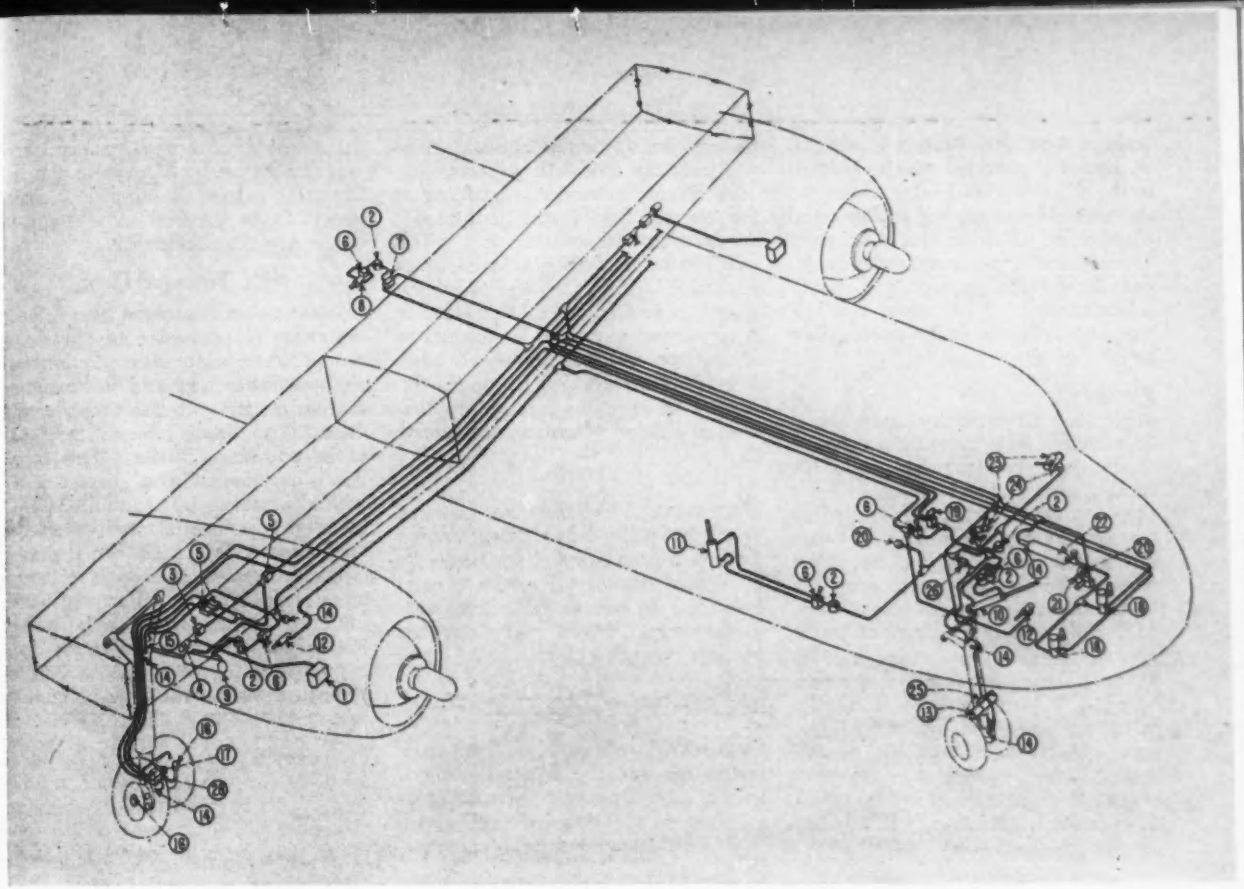


Fig 3—How the pneumatic system is installed. See Table I for component identification. A big factor in weight reduction is the exhaust of air to atmosphere; no return lines are required.

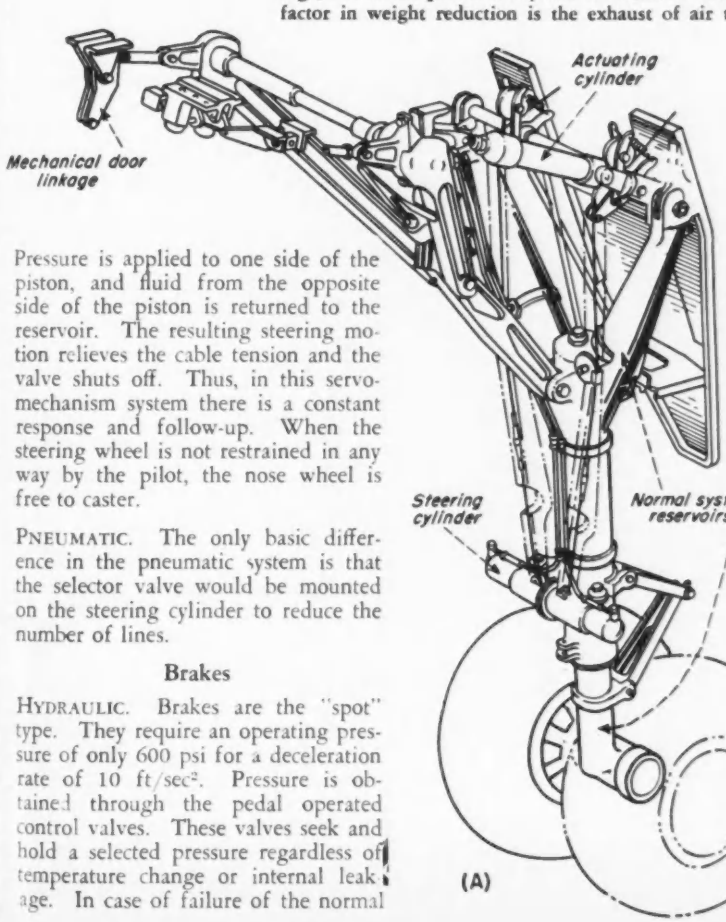
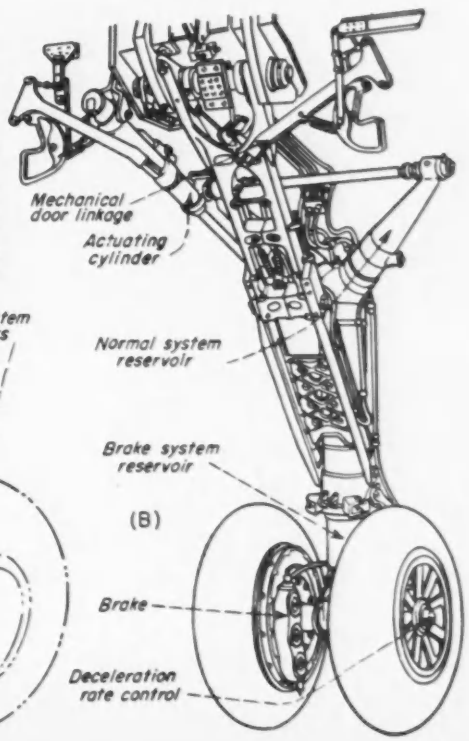


Fig. 4—Landing gear details. Cable operated selector valves are provided on each gear, and a viscous damper is built in the actuating cylinder to prevent excessive accelerations.



Pressure is applied to one side of the piston, and fluid from the opposite side of the piston is returned to the reservoir. The resulting steering motion relieves the cable tension and the valve shuts off. Thus, in this servomechanism system there is a constant response and follow-up. When the steering wheel is not restrained in any way by the pilot, the nose wheel is free to caster.

PNEUMATIC. The only basic difference in the pneumatic system is that the selector valve would be mounted on the steering cylinder to reduce the number of lines.

Brakes

HYDRAULIC. Brakes are the "spot" type. They require an operating pressure of only 600 psi for a deceleration rate of 10 ft/sec². Pressure is obtained through the pedal operated control valves. These valves seek and hold a selected pressure regardless of temperature change or internal leakage. In case of failure of the normal

brake system, an emergency pneumatic system is provided which sends air to the shuttle lockout valve, closes out the hydraulic system, and applies equal pressure to all four brakes at once. The amount of pressure can be modulated by the pilot through an "on-off" selector valve. After using this system, the hydraulic brake system must be bled of air.

PNEUMATIC. This system is the master and slave remote control type. Regulated braking pressure is supplied through the power valves from the landing gear piston reservoir. These power valves are remote controlled by the low pressure, pedal operated control valves (item 18, Fig. 3) which regulate the signal pressure from the brake control system air bottle (item 22). This permits instantaneous brake operation with a maximum of efficiency. The emergency or parking brake is applied through a shuttle valve that locks out the normal system. This system, like the normal brake system, provides differential braking and permits steering, which is a decided advantage. It receives its pressure from the normal system and pilot can modulate pressure.

A recent innovation pertaining to the aircraft field is the long-needed deceleration rate control unit. This unit can be used with a hydraulic or pneumatic system, but the pneumatic system is less complex. Its function is to prevent too rapid a deceleration of any one or all of the wheels when a sudden increase in friction coefficient, or of excessive pressure application, or lack of weight is encountered.

Flaps

HYDRAULIC. The flaps are powered by a 3,000 psi hydraulic motor driving through a gear box and positioned by a solenoid control valve with manual override. In case of failure, an auxiliary electric driven pump supplies pressure through a thermal relief check valve. The flaps are designed to extend or retract in about 15 seconds.

PNEUMATIC. A 400 psi pneumatic motor replaces the hydraulic motor and a cable operated valve replaces the solenoid operated valve. However, the solenoid valve with manual override may be used. A pressure regulating valve reduces the normal or emergency source pressure to 400 psi to obtain maximum efficiency.

Side Entrance Door

HYDRAULIC. The door assembly is designed to provide an overhead canopy when in the open position and the stairs fold down and out in simultaneous motion with the raising of the door. Operation is achieved by means of an actuating cylinder. This cylinder is positioned by a selector valve which is manually controlled and, when in the neutral position, locks fluid in the cylinder. When the door is open, excessive pressure due to external loads or thermal expansion is inhibited by means of a relief valve. Rate of raising and lowering the door is controlled by an adjustable thermal compensating restrictor valve and a balance-spring air cylinder.

PNEUMATICS. The pneumatic system actuation is similar. However, a pressure regulator is required to obtain maximum efficiency, and free fall provision must be achieved by means of viscous damping within the cylinder.

Cabin Pressurization

The present cabin pressurization system utilizes separately driven hydraulic pumps. The hydraulic fluid drives the compressor for the necessary air supply to the cabin. Replacement of the hydraulic portion of this system is possible when using an

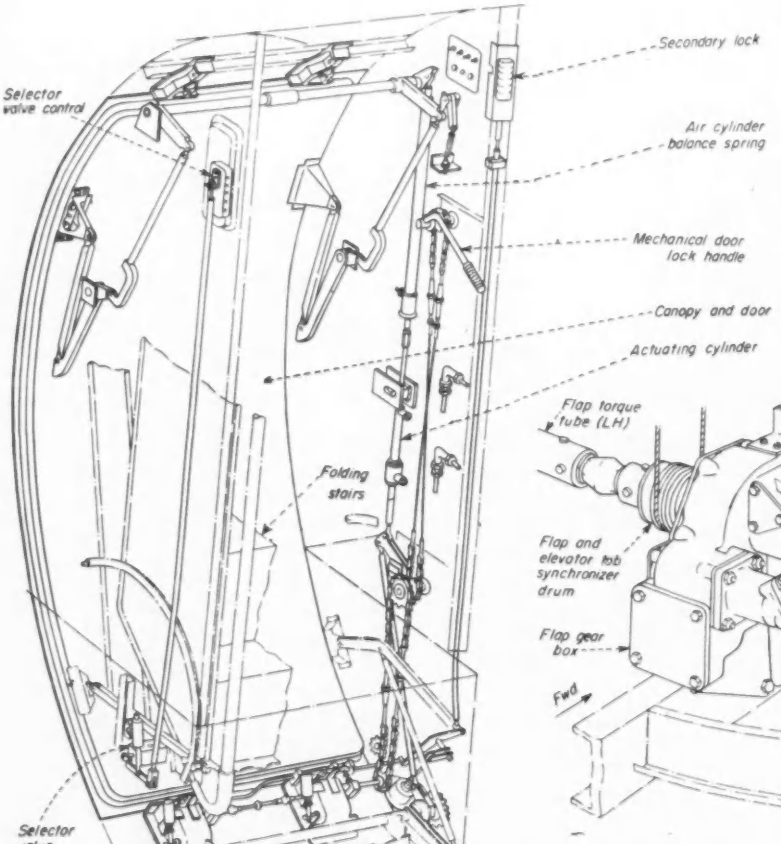


Fig. 5—Flap mechanism. Either a 3,000 psi hydraulic motor or 400 psi air motor can be used without making basic changes in the mechanism.

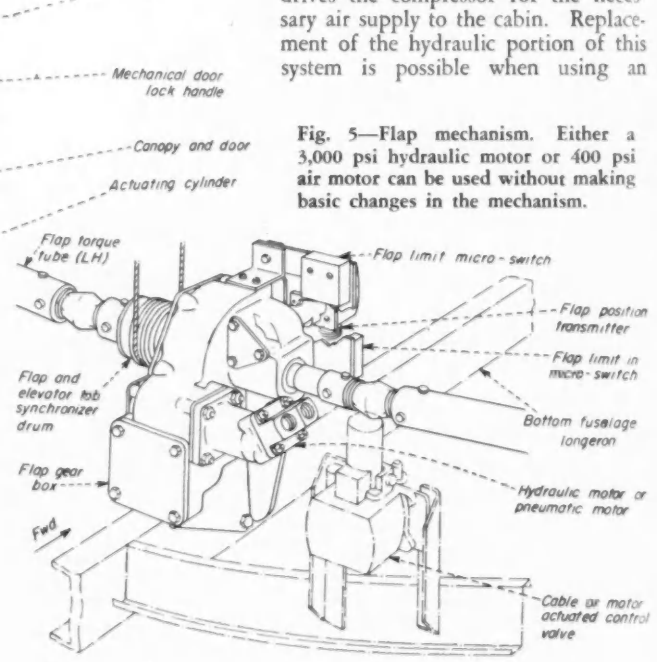


Fig. 6—The hydraulic and pneumatic mechanism on the side entrance door are similar. In the pneumatic system, a pressure regulator is used to achieve maximum efficiency, and viscous damping is provided in the actuating cylinder.

engine-driven compressor. This direct drive unit would mount on the present engine accessory pad, and air ducted directly to the cabin.

System Comparisons

A weight comparison of the hydraulic and pneumatic systems is given in Table I, including line sizes and lengths of tubing. Note that the weight of all components and piping for the pneumatic system is half that for the hydraulic system, and on aircraft, weight saving of this magnitude is extremely important.

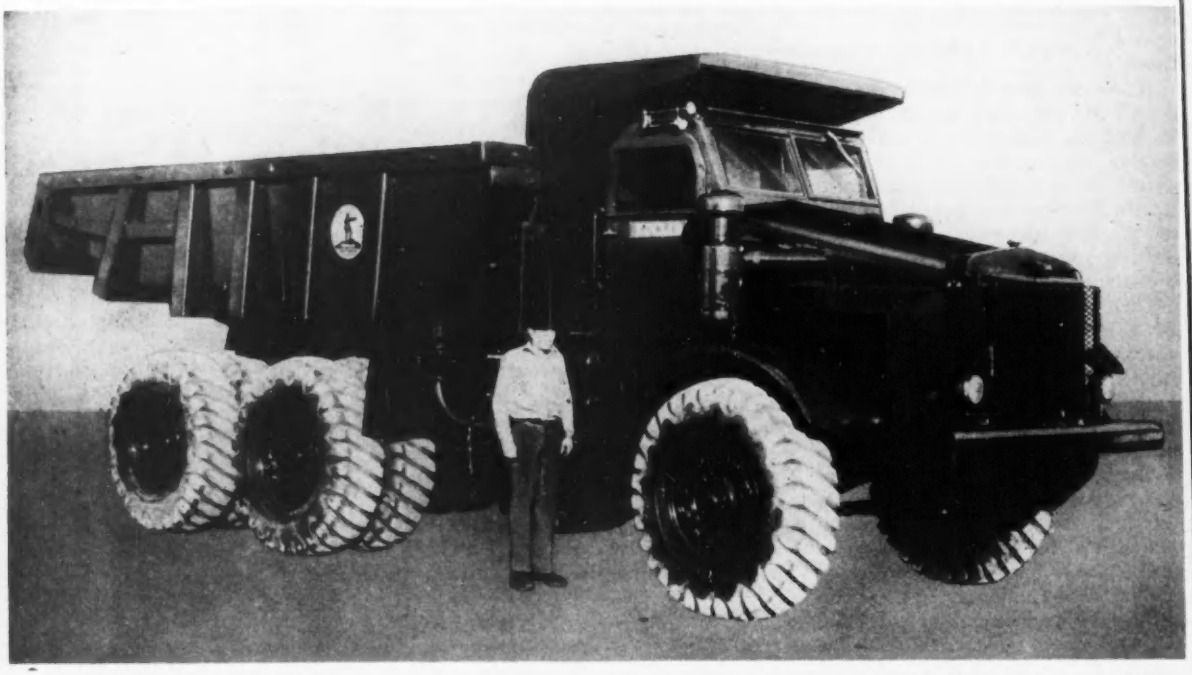
There is another major advantage to a pneumatic system, however; safety. A pneumatic system is fireproof. With present-day hydraulic fluids, a hydraulic system is not fireproof.

Another important item is the safety resulting from increased speed of operation of such functions as landing gear retraction. Single engine operation at take-off can be improved if the airplane drag can be decreased rapidly. The landing gear contributes considerably to the total take-off drag, and changes in gear retraction time should not be overlooked. The stored energy of the air is ready for instant use, and when released, split-second operation can be obtained. Exhausting the air to the atmosphere at the selector valve obviates the necessity for a return line since this air can be replaced readily while hydraulic fluid must be retained. For cold weather operation these return lines are required to be very large to provide for viscosity changes of hydraulic fluid. This factor is important when considering the weight of the system. Also, maintenance costs are less with a pneumatic system. Fewer lines and connections means less time required for inspection. Leakage from a hydraulic connection may spread over a large area of the airplane. Cleaning of these areas is a problem that does not exist with a pneumatic system. Detection of a hydraulic leak is accomplished by visible means whereas air lines can be checked by use of a soapy solution, as is done with oxygen systems, or by merely listening for the sound of escaping air. During any overhaul or routine maintenance work if an air-line must be disconnected the pressure should always be released. This is always done as a precautionary measure. Once the line is disconnected, replacement of a component can proceed. In the case of hydraulics, disconnecting a line requires equipment to catch the oil and also prevent its splashing over an area. The alternative is complete draining of the system which requires considerable time.

Table I—Comparative Weight of Hydraulic and Pneumatic Systems

System	Hydraulic units	No. units	Weight	Pneumatic units	Part No's (Fig. 3)	No. units	Weight
Power	Pumps—Engine Driven	2	13.8	Compressor—Eng. Driv.	1	2	37.0
	Elec. Driv.	1	14.7				
	Valves			Valves			
	Unloader	1	1.0	Unloader	3	2	4.6
	By-Pass	1	4.7	Relief	4	2	0.7
	Main Relief	1	1.4	Check	15	2	0.3
	Aux. Relief	1	0.9	Drain	14	9	.9
	Shut-Off	2	2.5	By-Pass	5	6	1.8
	Ground Service	1	0.7	Ground Service	21	1	.4
	Check	7	2.0				
	Reservoir	1	17.5	Reservoir	27	3	2.5
	Accumulator	1	14.3	Rework of the			
	Noise Filter	2	4.2	Landing Gear			
	Gauges	3	1.5	Fuse	26	1	0.4
	Gauge Transmitters	2	1.0	Gauges	20	2	1.0
		26	80.2			30	49.6
Landing Gear	Cylinders			Cylinders			
	Main Gear	2	22.4	Main Gear	9	2	31.0
	Nose Gear	1	6.0	Nose Gear	10	1	9.0
	Unlatch	3	2.3	Unlatch	12	3	2.3
	Valves			Valves			
	Up and Down	1	1.2	Up and Down	6	3	5.3
	Unlatch	1	0.5	Unlatch	6	1	0.5
	Relief	3	3.4	Relief	4	1	0.4
	Check	12	3.0	Press. Regul.	2	3	4.5
		23	38.8			14	53.0
Flaps	Motor	1	1.6	Motor	8	1	3.0
	Valves			Valves			
	Up and Down	1	4.0	Up and Down	6	1	2.0
	Thermal Rel.	1	0.3	Press. Reg.	2	1	1.5
				Shuttle	7	1	0.2
		3	5.9			4	6.7
Brakes	Valves			Valves			
	Power	2	4.2	Control	18	2	2.2
	Shuttle and Lock-Out	2	5.0	Power	28	2	3.0
	Emerg. Air	1	0.7	Shuttle	17	2	1.3
	Air Bottle	1	3.9	Emerg. or Park	19	2	2.0
	Gauge	1	0.5	Control Air Bottle	22	1	6.0
				Deceleration Rate	16		
				Control Unit[a, b]			
		7	14.3			9	14.5
Nose Wheel Steering	Cylinder	1	15.0	Cylinder	13	1	13.0
	Valves			Valves			
	Control	1	1.8	Control	25	1	0.6
	Check	2	0.5				
		4	17.3			2	13.6
Side Entrance Door	Cylinder	1	7.8	Cylinder	11	1	10.0
	Valves			Valves			
	Up and Down	1	0.7	Up and Down	6	1	0.6
	Relief	1	0.9	Press. Reg.	2	1	1.5
	Check	3	0.7				
	Adj. Restr.	1	0.4				
		7	10.5			3	12.1
Windshield Wipers	Wiper Units	2	5.0	Wiper Units	23	2	5.0
	Valves			Valves			
	Speed Cont.	1	0.3	Speed Cont.	24	2	0.5
	Regulator	1	2.5	Press. Reg.	2	1	1.5
	Fuse	1	0.9				
		5	8.7			5	7.0
General	Tubing and Fittings		143.7	Tubing and Fittings			26.0
	Sizes 1.25 O. D.			Size 0.25 O. D. Only			
	1.00 O. D.						
	0.75 O. D.						
	0.50 O. D.						
	0.38 O. D.						
	0.25 O. D.						
	0.19 O. D.						
	Fluid		70.0	Fluid			5.0
	Brackets, Tube Clamps and Miscellaneous		36.4	Brackets, Tube Clamps and Miscellaneous			25.0
			250.1				56.0
	Grand Total		425.8				212.5

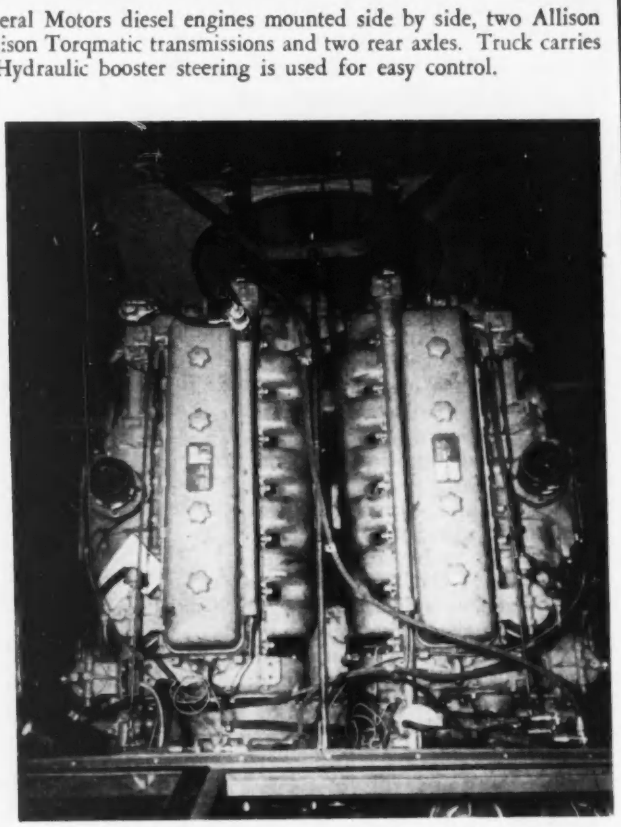
Weight difference, or weight saved:
(a) Without deceleration rate control unit in pneumatic system—213.30 lbs.
(b) With deceleration unit in pneumatic system —131.30 lbs.



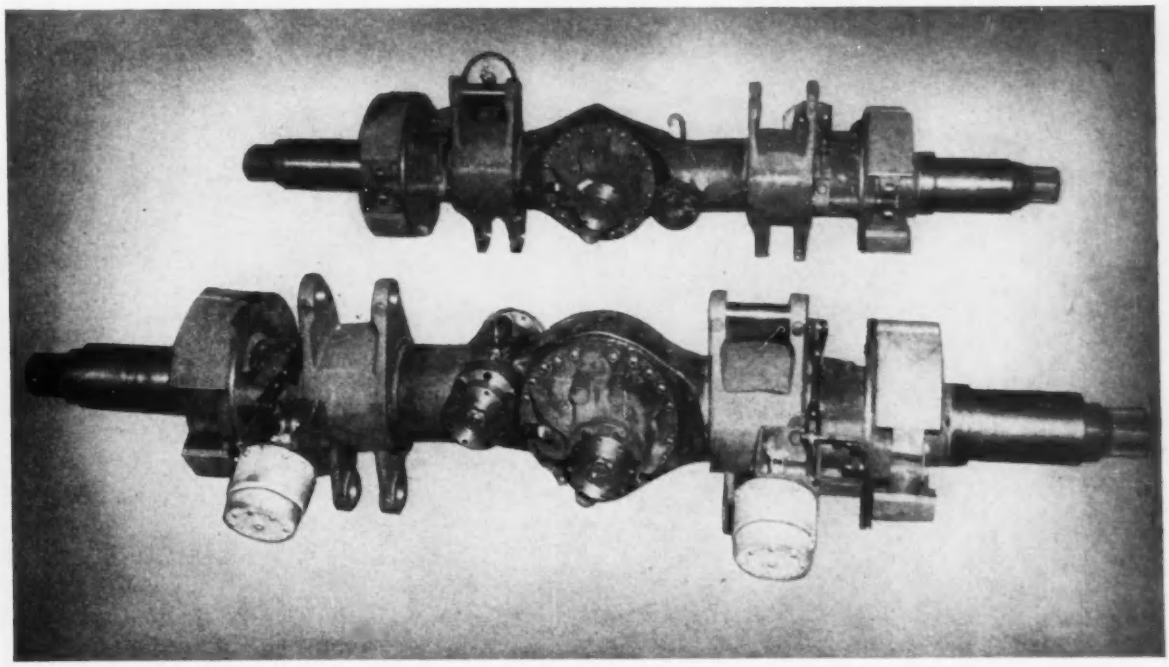
HEAVY DUTY DUMP TRUCK has two 190 hp General Motors diesel engines mounted side by side, two Allison torque converters mounted integral with engines, two Allison Torqmatic transmissions and two rear axles. Truck carries a payload of 34 tons at a top speed exceeding 26 mph. Hydraulic booster steering is used for easy control.

Torque Converters Eliminate Use Of Power Divider

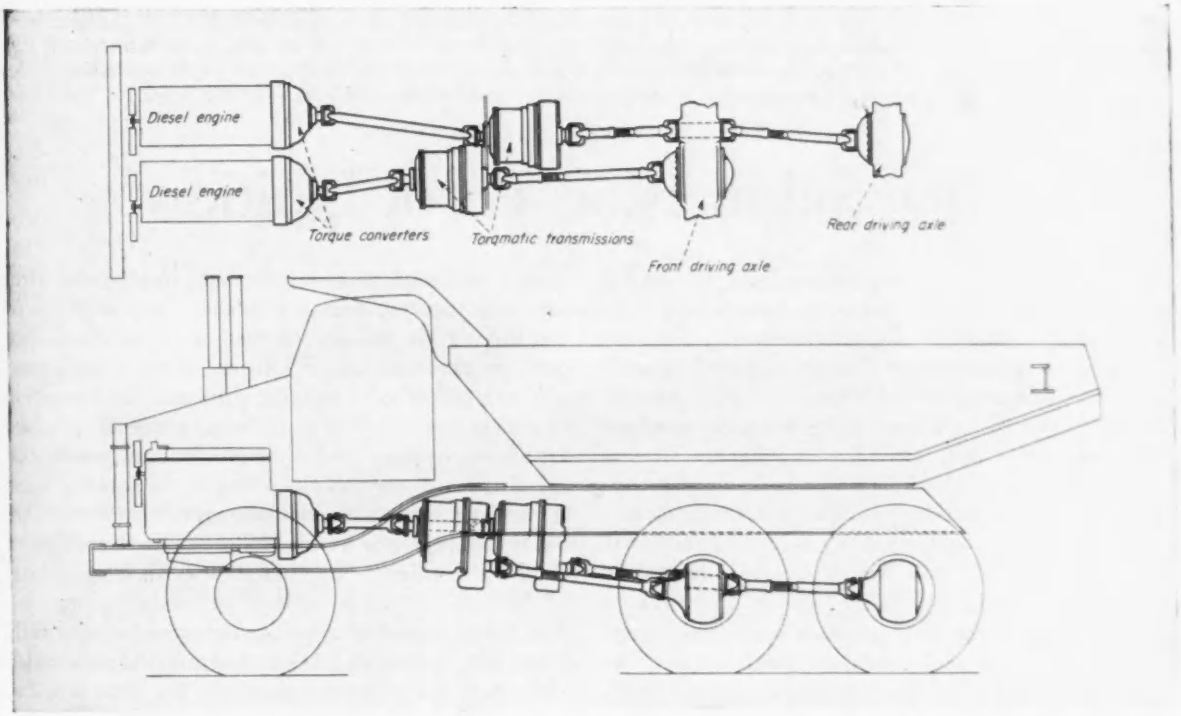
When the Euclid Road Machinery Co., of Cleveland, Ohio, put two diesel engines in their model FFD dump truck, the two power sources raised a common problem in power division and synchronization. The conventional solution would be an interaxle power divider. Another way, the Euclid choice, was to incorporate a torque converter with each diesel engine, and use two Allison Torqumatic transmissions and two rear axles. Each engine, therefore, has its own torque converter, transmission, and axle. The two rear axles are of the double reduction type, with single reduction differential and final reductions in each wheel consisting of floating planetary systems. As a result of this power train, there is no clutch pedal or manual gear shift. The operator can change gears under full power at any speed.



POWER PLANT consists of two General Motors 190 hp diesels. They are six cylinder engines developing 1,200 ft-lb torque at 1,200 rpm, and weighing 2,400 lb each. The Allison torque converter mounted integral with each engine has a 3.6:1 max torque multiplication.



TWO POWER AXLES operate independently of each other and are spring mounted about a central pivot point to produce a bogey action. The drive line for the rearmost axle passes through a shaft in the first rear axle. The planetary final reduction between the differential and wheel has a reduction of 4.58:1, to give a total of 14.2:1 reduction.

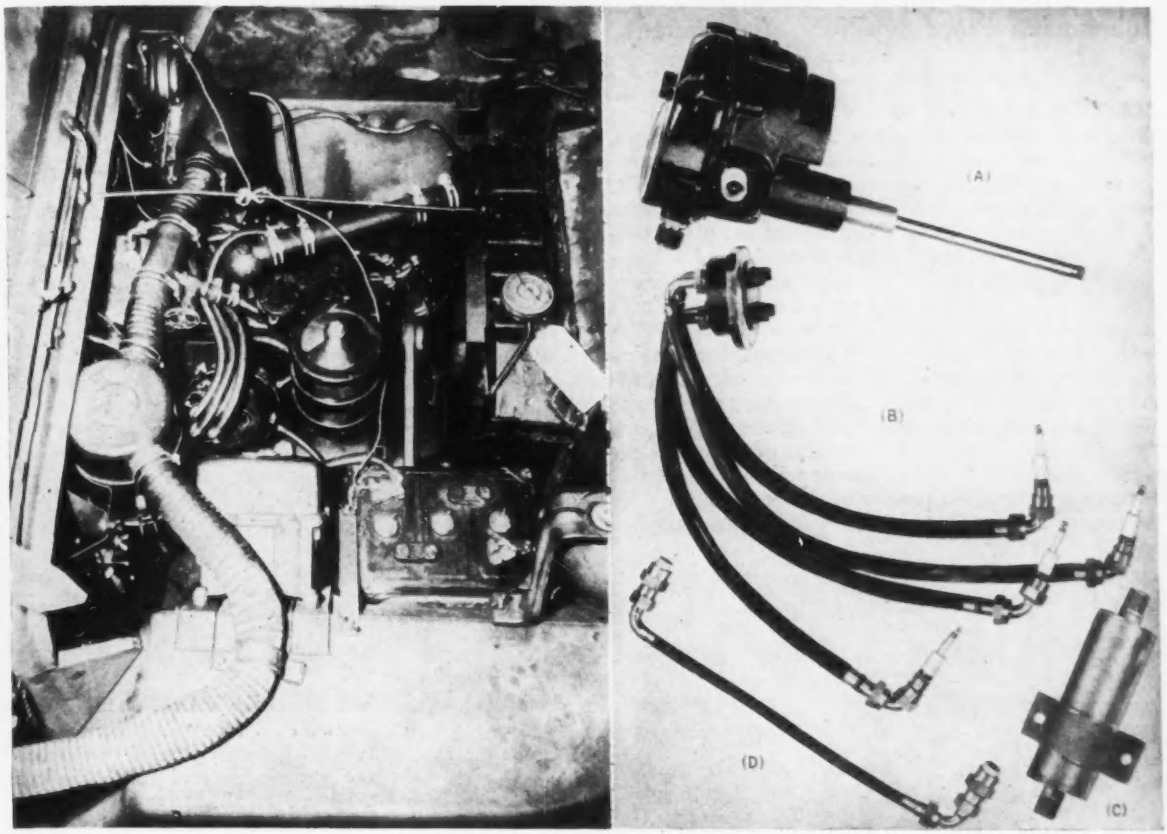


COMPLETE POWER TRAIN is shown in schematic drawing. Notice how the two engines work completely independently through two complete drive systems. The torque converters eliminate the conventional inter-axle power divider. The two transmissions have planetary gearing, with three forward speeds and one reverse. They are controlled by a mechanical linkage to a hydraulic selector valve. Transmissions are mounted rigid to frame side members.

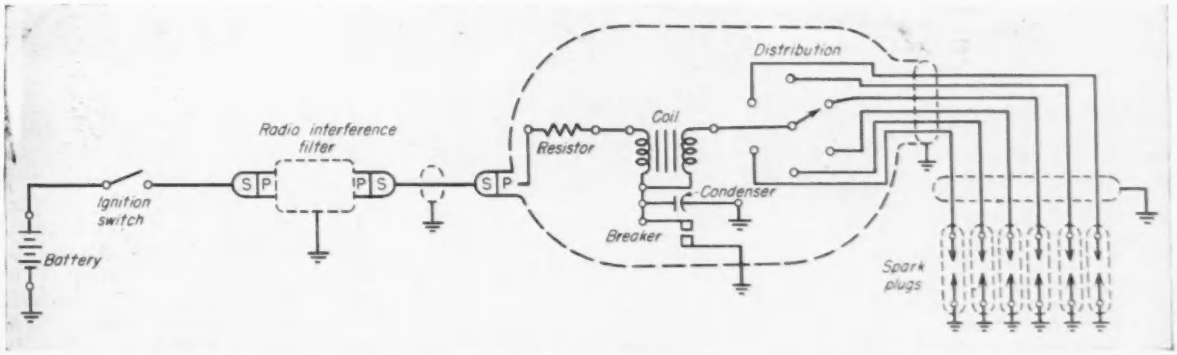
FIRST APPLICATION of the new submersion-proof system was on the Navy's underwater Jeep—the "Snorkel"—made by Willys Overland Motors, Inc. Bendix engineers are predicting widespread use on such equipment as military vehicles, marine and amphibious craft, shipboard generator equipment for starting jet aircraft, and commercial trucks and buses. Tests aggregating over half a million miles have been conducted by several of the larger bus operators.

New Submersion Proof Ignition

Waterproof battery ignition systems used in gasoline engines and particularly automobiles have been in existence for a number of years. However, they were not designed to be completely submersion proof. Usually they were designed for operation only under periodic, intermitten spray conditions. That's where the new ignition system made by the Scintilla Magneto Div. of Bendix Aviation Corp. differs. It was designed for the worst possible condition—underwater operation. It is capable of being submerged in salt water and will exclude this water even when subjected to an internal vacuum of 6 psi. It is a radio shielded unit and will meet Army, Navy, and Air Force specifications in this respect, as well as being corrosion resistant, fungi resistant, and meeting high and low temperature requirements. Obtaining features of this type meant engineering the system from the ground up. The distributor housing is an aluminum alloy die casting especially impregnated to prevent moisture seepage. All leads, including the high tension leads to the spark plugs and the radio interference filter lead, are waterproofed to eliminate the possibility of shorting out to ground. Ventilation of the distributor prevents accumulation of nitrous oxide fumes, thus avoiding corrosion of metallic parts and the formation of current carrying films on dielectric parts. All external fittings are stainless steel. A metallic bellows assembly on the distributor provides for oiling of the internal cam. A positive resistance in series with the coil prevents burnout if the ignition is left on and the contacts are closed. Since the distributor "bug" or rotor is an integral part of the cam, the cam is made of an insulating material. And finally, the advance unit in the timer has unusually heavy duty flyweights fabricated of sintered iron which were found less costly than forgings. The same distributor housing and cover can be used for either four or six-cylinder systems, and the distributor can be simply altered for either direction of rotation. A different housing is required for 8-cylinder units.



CLOSE-UP OF THE JEEP "Snorkel" powerplant. Spark plugs are of the shielded type used on aircraft. Main electrical elements of the ignition system are: Distributor (*A*); high tension leads (*B*); filter (*C*); and primary lead (*D*). Two types of high tension leads are available, one for unshielded and one for radio shielded installations; both are waterproof. Distributor encloses conventional cam actuated breaker points and revolving shaft contact arm.

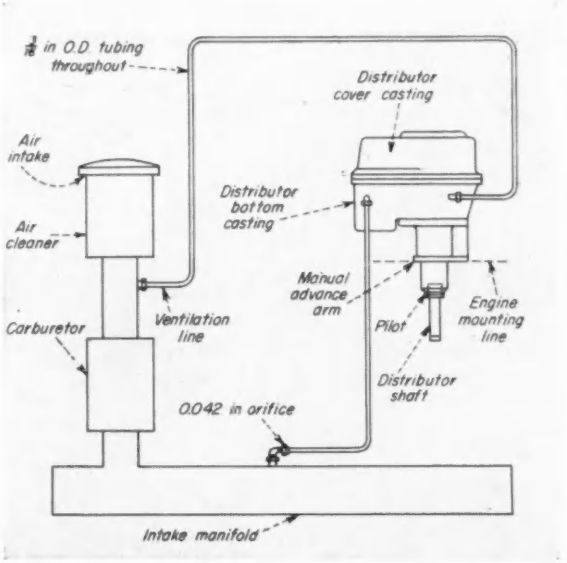


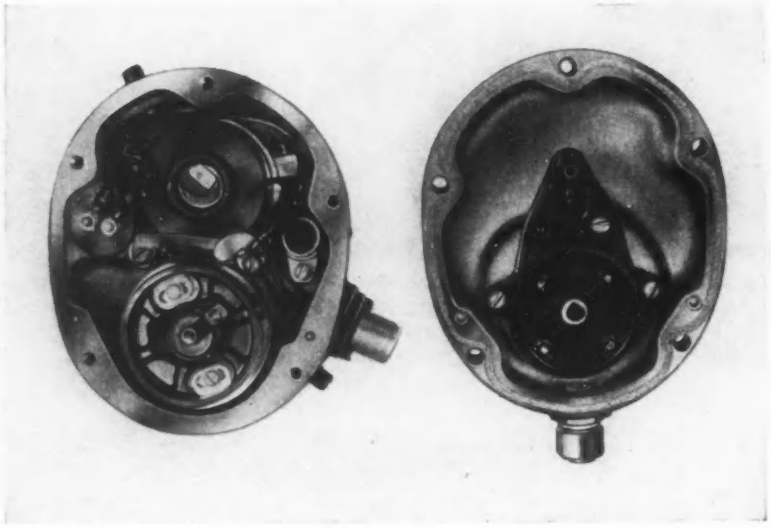
CURRENT FROM THE BATTERY enters the timer through the electrical connector on the main housing and flows through the control resistor, the primary winding of the coil and the breaker contact to ground. The coil is essentially a step-up transformer with two windings, a primary and a secondary, on a laminated silicon iron core. The secondary is wound with a much larger number of turns than the primary. With the contact breaker closed and the current flowing in the primary, a strong magnetic field is built up in the coil. When the breaker opens, this field collapses almost instantly, causing a high voltage to be induced in the secondary winding. The voltage in the secondary rises until a value is reached that will break down the gap in the spark plug. The voltage now drops abruptly and the remaining energy pours across the spark plug electrodes in the form of a strong, hot spark. Since the maximum distributor output is considerably greater than that needed to break down the spark plug gap, a large reserve is available for this purpose. The high voltage impulse from the secondary coil winding goes to the distributor finger and is delivered to one of the electrodes in the distributor block where it is conducted to the spark plug. The breaker contact is closed for a comparatively long period to allow the magnetic field to reach a maximum.

(continued on next page)

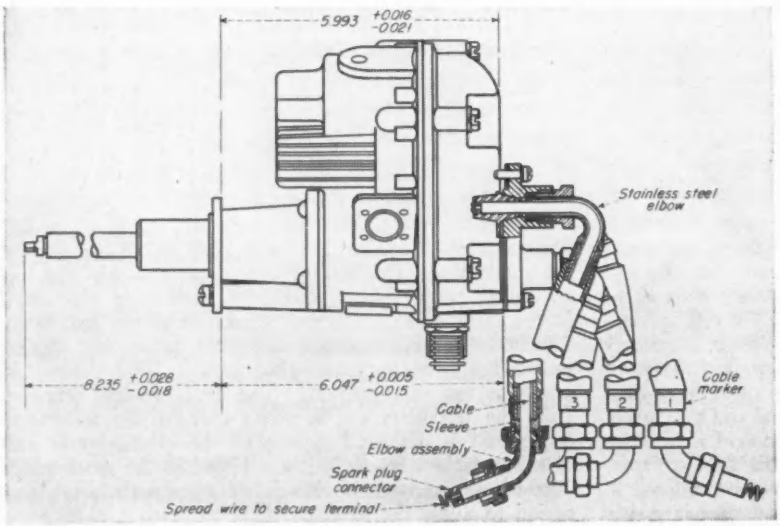
Submersion Proof Ignition (continued)

MAIN DISTRIBUTOR HOUSING (below) contains a drive shaft, advance mechanism, breaker cam and finger, coil, condenser, control resistor and breaker. The distributor block and high tension outlets are in the cover. The distributor is driven at one-half engine speed. The drive shaft is supported by one ball bearing and one plain bushing. The plain bushing is on the drive end and is lubricated by splash from the engine. The distributor finger mounts on the advance mechanism and transmits the high tension impulse from the coil to the distributor block electrodes. The finger is also used to operate the breaker contact by means of the cam lobes on its outer edge. The advance mechanism is located in the main housing below the distributor finger and is used to provide the proper spark timing for any engine speed. Operation of the unit is entirely automatic. The control resistor provides a means of dissipating externally a portion of the heat generated by the current in the primary circuit with contacts closed.





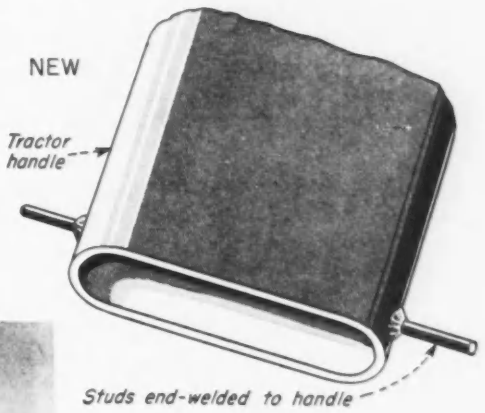
METHOD OF VENTILATING DISTRIBUTOR (above). Forced ventilation is absolutely necessary to remove the undesirable acid forming oxides produced in all equipment where an electric spark is formed, and thus prevent corrosion of internal parts. A ⅜ in. copper tubing line is connected from the carburetor air filter to the timer and a second line from the timer to the engine intake manifold. An orifice in the line controls air flow.

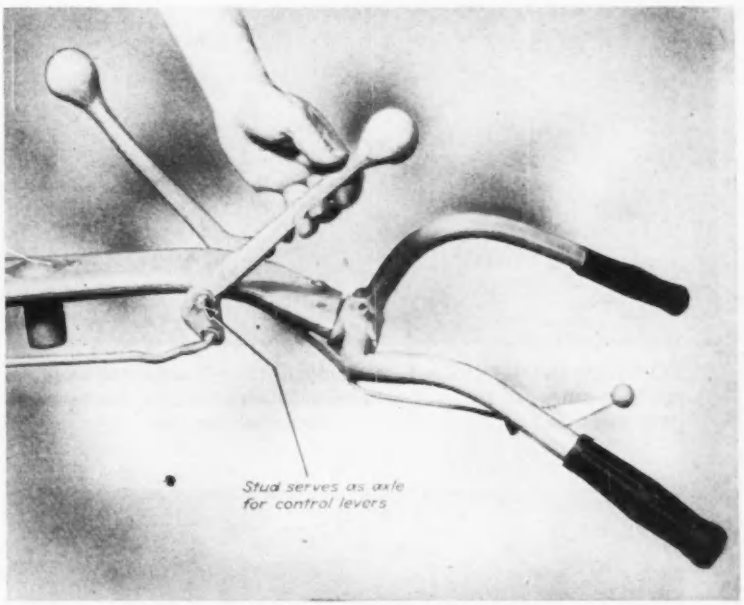


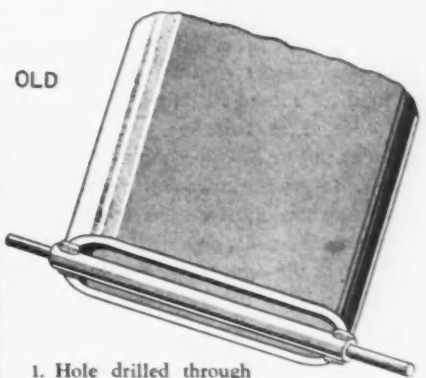
HOUSING AND CONDUIT waterproofing details. Die-cast aluminum alloy housing is finished with zinc chromate primer and two coats of black baked enamel. A brass bellows, external means of oiling the cam, is brazed to a stainless steel fitting. This unit is made waterproof in the housing by a soft aluminum washer which forms a seal when the fitting is tightened down. Spark plug connections are waterproofed by either a soft copper washer or a spherical seat connection. Cover is moisture-proofed with a neoprene gasket of circular cross section that has a braided cotton thread and aluminum wire covering. Aluminum allows electrical contact while the thread is treated with a sealing compound. The sinteres iron rotor flyweights are held at 90 percent minimum density.

Stud Welding Cuts Costs In Tractor Handle Assembly

When the Midland Co., of South Milwaukee, Wis. began looking for faster and cheaper methods to produce their garden tractor, stud welding proved to be at least one answer to their problems. The company now uses the stud welding process extensively to effect product economies with no sacrifice in quality. A few examples of this extensive use of the stud welding process are shown in illustrations below. In most cases, significant weight savings were an extra dividend.



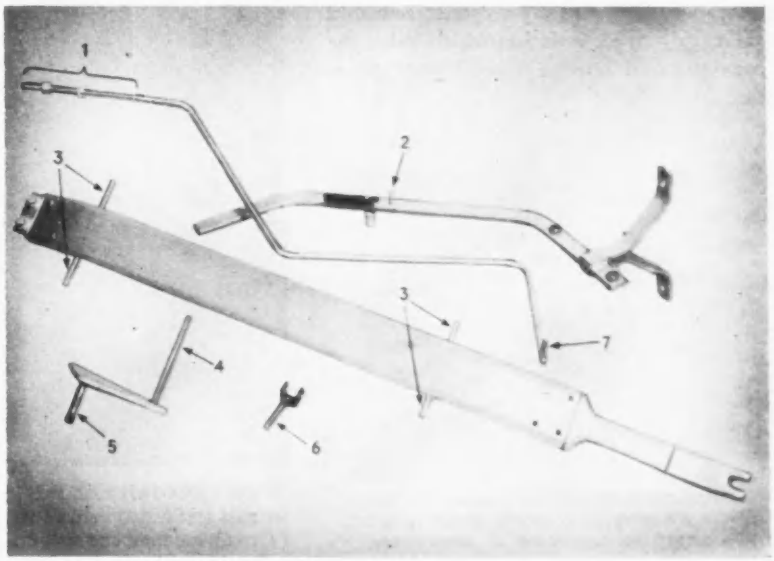




1. Hole drilled through handle
2. Hollow tube inserted and hand welded in place
3. Axle rod inserted in hollow tube

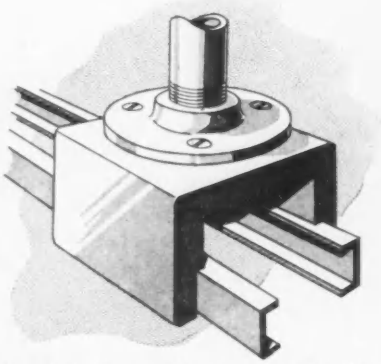
IN TRACTOR HANDLE the end welded studs serve as axles for the control rod levers. Under the old method, (right, above) holes were drilled in the tractor handle and a hollow tube was inserted in the holes and hand welded into place. A solid rod was then inserted in the tube to form an axle. Nelson Stud Welding equipment is used throughout.

VARIOUS STUD WELDING applications are: (1) A 6 in. stud that gives better adjustment of the control rod; (2) stud acts as stop pin; (3) axles for control rod levers; (4) a cultivator handle; (5) cultivator axle; (6) a connecting rod yoke—weight and cost were reduced by 50 percent; and (7) a stud with a cross-drilled hole. The stud eliminates a previously used expensive bending operation.

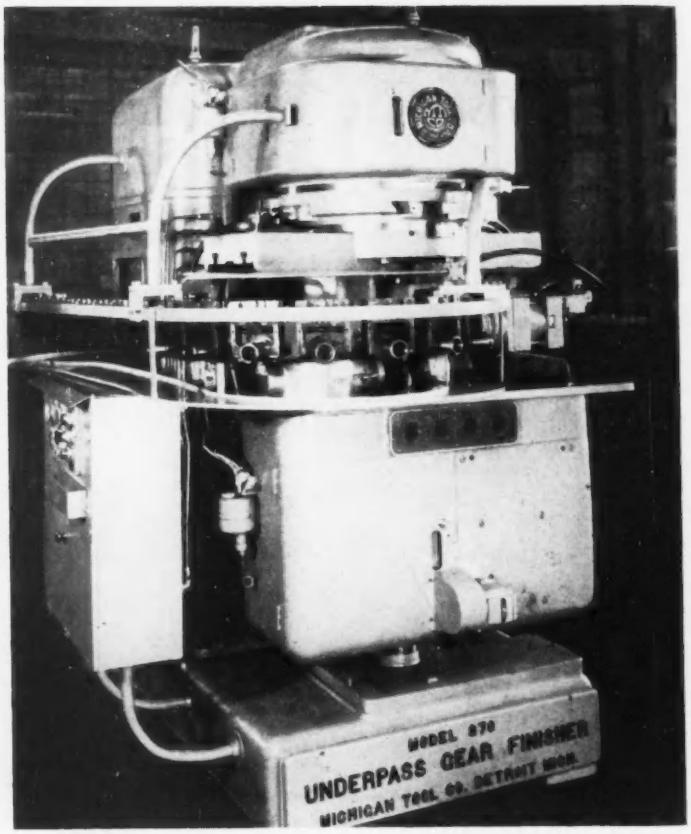


Conveyor System For Automatic Gear Finisher

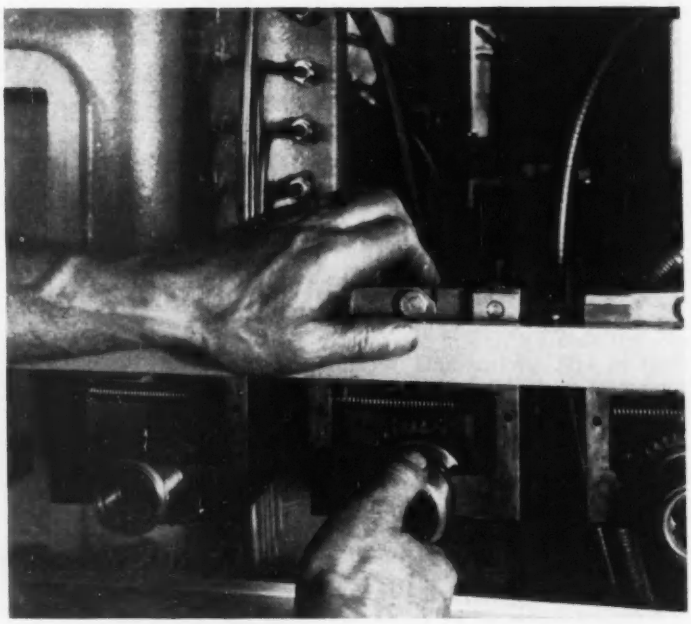
The application of a loop transfermatic conveyor to one or more gear finishing machines provides continuous operation at peak efficiency. The conveyor application is made possible by the new "arch way" column type of machine made by the Michigan Tool Company of Detroit, Mich., in which the gears to be machined are carried by small "carts" coupled together to form a continuous conveyor. Each cart has jaws that carry the gear for which they were designed. Finger pressure opens the jaws to load or unload the carts. The operating sequence is as follows: a horizontal air cylinder located above the conveyor and inside the column advances the entire conveyor one station at a time. After it stops, air operated centers engage the gear. When clamped in the centers the cart fingers open up and release the gear for the shaving process. When the cutting cycle is complete, the cart jaws pick up the gear again, the centers retract and the conveyor is indexed to the next station. The operating cycle requires about 20 seconds from "floor to floor." The indexing from one station to another, engaging centers and releasing the cart jaws only takes 2.5 seconds. Solenoid valves govern air cylinders which index the conveyor and operate pickup mechanism.



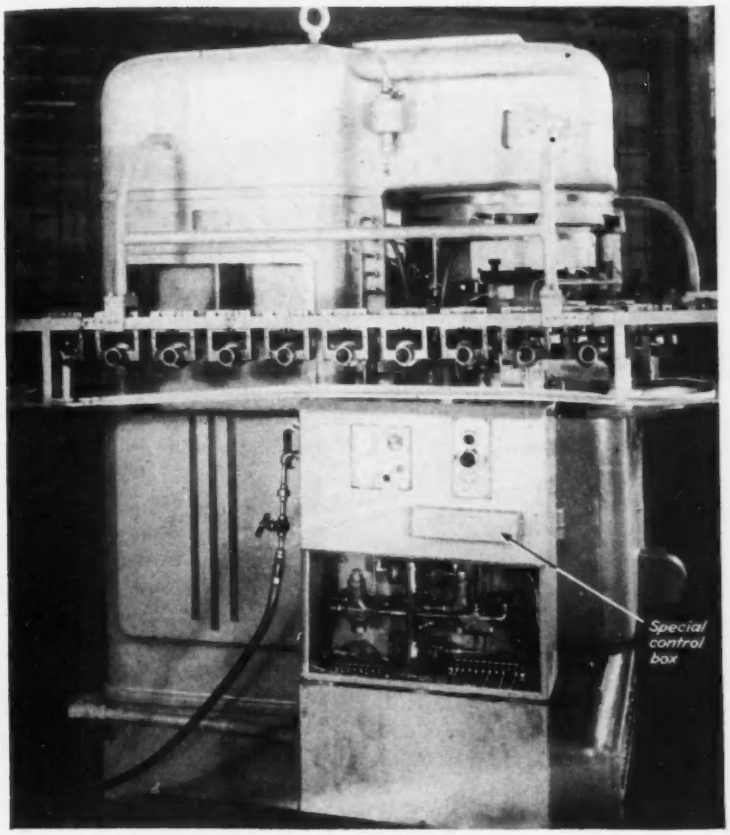
Conveyor tracks are strongly supported, above. Tracks are made of standard rolled channel, with curved sections made of boiler plate.



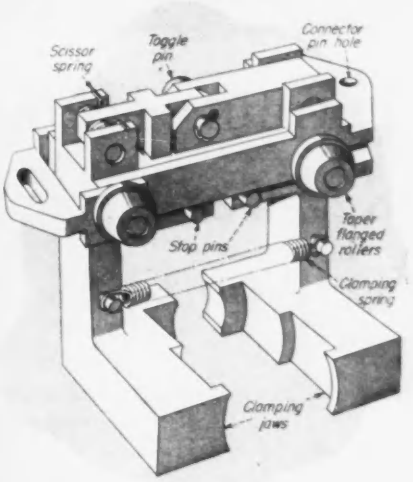
TO INSURE ALIGNMENT, the conveyor track is supported from the machine, instead of from the floor. Controls are placed at the operators position instead of the usual place in front of the machine.



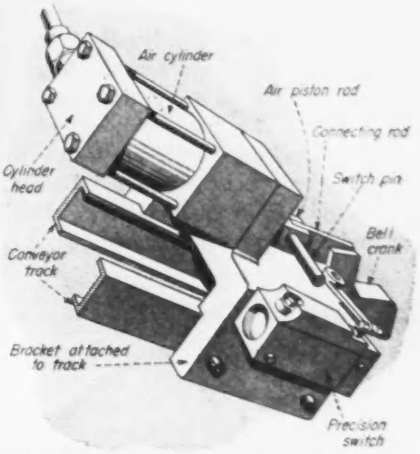
VIEW FROM OPERATOR'S STATION shows air supply line, control buttons and air control valves inside the open cabinet. Special controls (above) are used for setting up and adjusting the gear finisher.



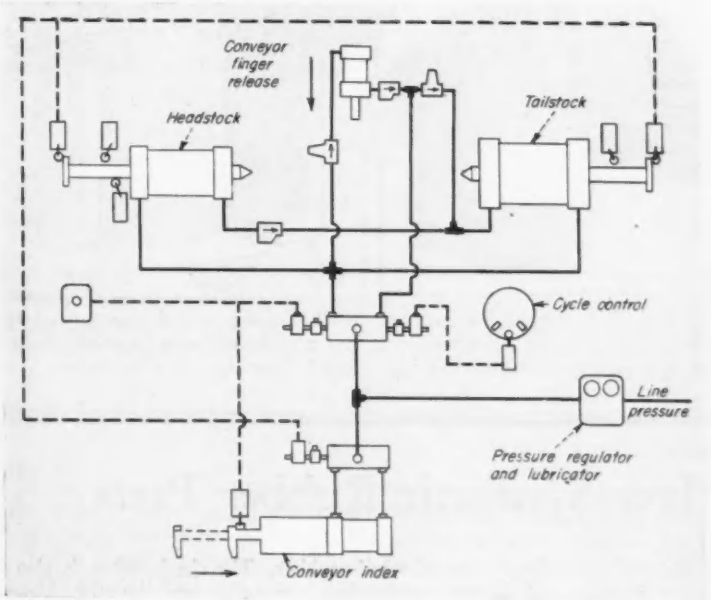
OPERATOR HAS JUST INSERTED a gear in its "cart", and the arms have closed on the piece. Light finger pressure at the center opens the fingers which hold the gear. Flanged tapered wheels support the cart.



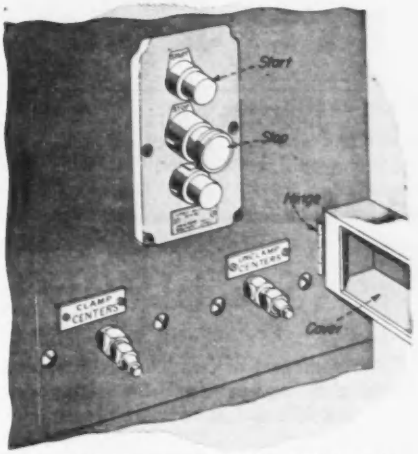
Detail of one gear carrier "cart" shows the relation of the swinging clamping arms to the carriage. Big return spring is easily released.



Clamping arms are released by air-operated bell crank when centers have taken over the work. Cylinder is bracket mounted.



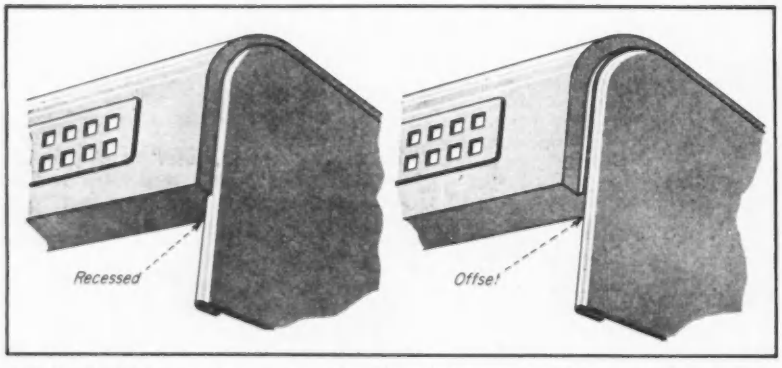
AIR LINE DIAGRAM shows main operating units. In operation, the tailstock moves forward and stops, the headstock moves forward and clamps, and then the conveyor fingers release. After shaving cycle the machine stops while fingers close and head and tail stock retract.



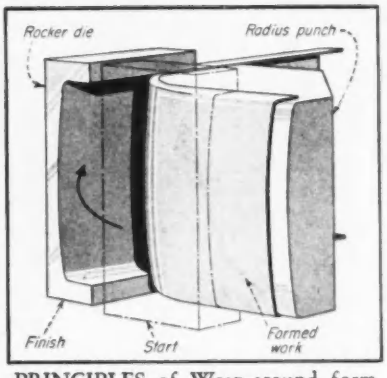
Special controls are provided beneath the regular control buttons for setting up and adjusting the gear shaver.

Forming Method Reduces Heater Tooling Cost

Since the "Heat-Aire" space heater was to be produced in the medium quantity range, its rectangular design presented a problem in tooling costs that ruled out the use of expensive dies and stamping presses. Using the comparatively new "tangent bending", or "Wrap-around forming" process, machine and die costs were brought to a reasonable figure with no sacrifice in appearance. The process is usually called "Tangent bending" because the forming action of the work takes place tangentially to the form punch or die. Only the male punch has any form or development work. A rocking female die forms the work around the punch. The Cyril Bath Co., of Cleveland, Ohio, manufactures the Wrap-around forming equipment used on heater.



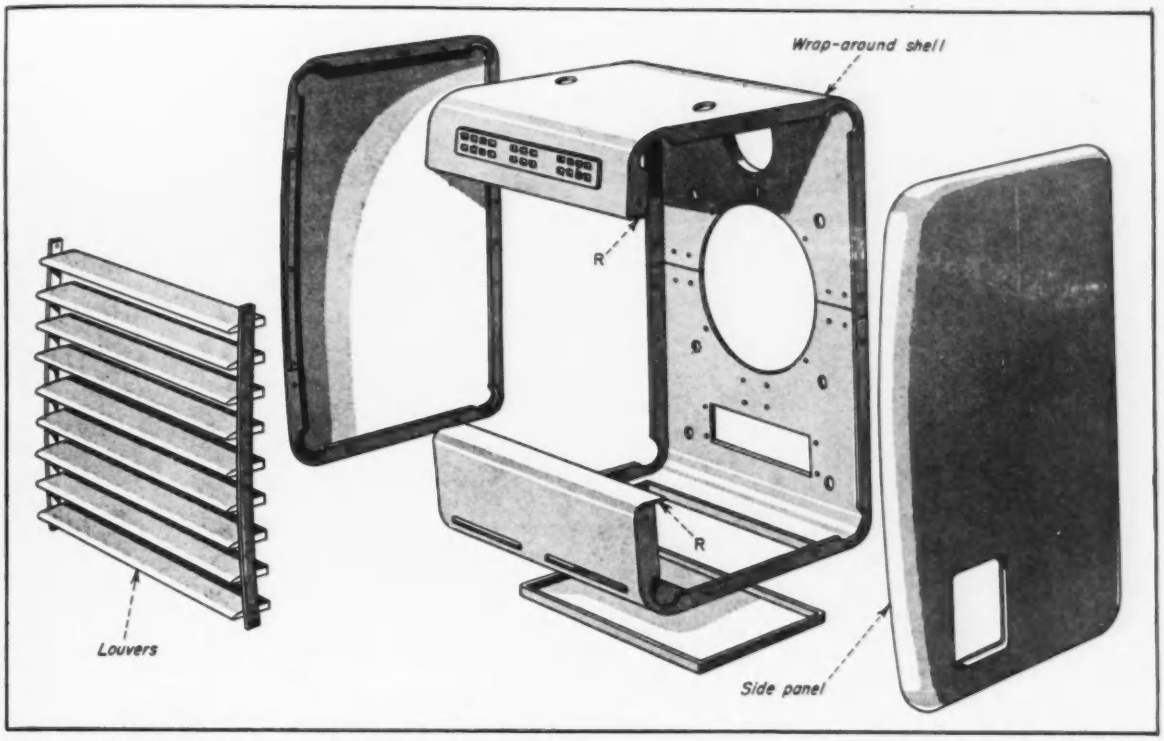
SIDE PANELS are stampings. If a flat plate were used, one of the two approaches, above, could be followed. Both cases call for a restrike operation. Either would be cheaper, but with a poorer appearance, than a stamping.



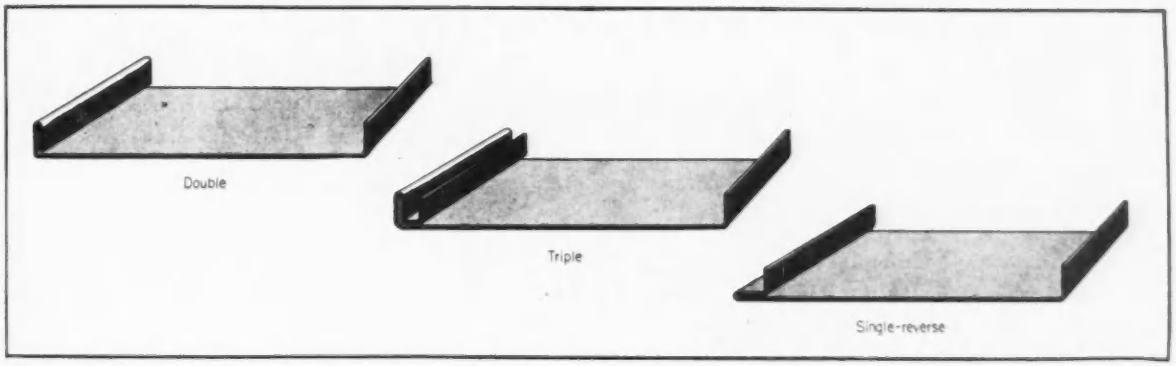
PRINCIPLES of Wrap-around forming are shown as a preformed sheet metal channel is bent to shape.

Special Valves Have Synthetic Rubber Parts

When engineers of Bendix Home Appliances, Inc., substituted the "squeeze-dry" for "spin-dry" feature in the automatic "Economat" washing machine the required components included two specially designed valves—the vacuum release and check. The successful performance of the vacuum release depends on the quality of several valve parts: a diaphragm, valve seat, and valve seal, all molded from "Hycar" nitrile rubber. The check valve, is also an all-rubber construction. Manufactured by the Vermay Laboratories, Inc., of Yellow Springs, Ohio, these parts have been tailormade to Bendix specifications and are examples of precision engineering and molding in rubber. All four parts are molded to close tolerances and must resist dimensional change during the life of the machine



WRAP-AROUND SHELL forms the top, back, and the bottom and part of the front of this heater case. Before the four radius bends were formed in this piece, the channel was preformed with all necessary notches, cutouts, and return flanges. The return flanges, marked "R", were welded and ground before the wrap-around operation.



DOUBLE FLANGES, for strength, and triple flanges, for both strength and insertion features, can be formed by the wrap-around method. It has been found that 20 gage metal is the most desirable thickness to form in steel, with 22 gage being the lightest material that will allow a smooth flange to be formed wrinkle-free.

to insure successful performance in home operation.

The Bendix Economat employs agitator washing action, but in rinsing and water extraction departs from the spin dry feature found in other automatics. A flexible tub takes the place of the metal one more commonly used, and by means of a vacuum pump both water and air inside the tub are removed. Atmospheric pressure on the outside of the tub then causes it to collapse, pressing the water from the clothes. Timed to work in conjunction with the vacuum pump is the vacuum release valve, operated not by a conventional solenoid but by line water pressure. After the squeeze phase, during which this valve is held shut by vacuum in the machine, water pressure opens the valve to fill the unit for rinsing.

Because these synthetic rubber parts must fit precisely in the valve assembly, so that positive and dependable operation is assured in thousands of washing cycles, it is required that they possess low water absorption, low compression set, and resistance to fluid aging. The check valve must also be resistant to alkalies, soaps, detergents, and bleaches.

Table I—General Properties of Commercial Synthetic and Natural Rubbers

Classification	Oil-Resistant Type			Non-oil-Resistant Type		Nat. Rub. Ref. Std.
Elastomeric polymer	Butadiene acrylonitrile	Neoprene	Thiokol	Butadiene Styrene	Butyl	Natural Rubber
Government designation	GR-A	GR-M	GR-P	GR-S	GR-I	
Properties						
Vulcanization method	Heat and sulfur	Heat alone	Heat alone	Heat and sulfur	Heat and sulfur	Heat and sulfur
Odor	Poor	Poor	Bad	Fair	Excel.	Good
Efficiency						
Rebound, hot	Good	Good	Poor	Good	Good	Excel.
Rebound, cold	Fair to good	Fair to good	Poor	Fair to good	Very poor	Excel.
Speed of recovery	Poor	Fair	Very poor	Good	Very poor	Excel.
Tear resistance	Fair	Fair to good	Fair	Poor to fair	Good	Good
Abrasion resistance	Good	Excel.	Poor	Good	Good	Excel.
Aging resistance in:						
Sunlight (ozone)	Poor	Excel.	Good	Poor	Excel.	Poor
Oxidation (oxygen)	Fair	Good	Good	Good	Good	Good
Heat	Good	Very good	Poor	Fair	Good	Fair
Static (shelf aging)	Good	Good	Good	Good	Good	Fair
Resistance to flex cracking						
Extension flexing	Fair	Good	Poor	Poor to fair	Good	Good
Bending flexing	Good	Good	Poor	Good	Very good	Good
Resistance to comp. set (or permanent set)	High	Medium	Very low	Medium	Medium	High
Solvent resistance						
Hydrocarbon	Good	Fair to good except aromatics	Excel.	Poor	Poor	Poor
Oxygenated solvents	Poor	Poor	Good	Good	Good	Good
Oil and gasoline resistance	Excel.	Good	Excel.	Poor to fair	Poor	Poor
Acid resistance	Good	Excel.	Fair	Good	Excel.	Excel.
Flame resistance	Poor	Good	Poor	Poor	Poor	Poor
Cold resistance (freezing or hardening on cooling)	Fair	Poor	Poor	Fair to good	Fair	Good
Permeability to gases	Medium	Low	Low	Medium	Extra low	Medium
Volume resistivity ohm/cc. at 25 C	10^{-10}	10^{-11}	10^{-13}	10^{-15}	10^{-17}	10^{-12}—10^{-1}
Dielectric constant at 1000 cycles	13.0	9.0	6.0	2.9	2.4	2.7
Power factor at 1000 cycles	5.5	3.0	....	0.32	0.3	0.23
Resistance to hot-water swelling	Excel.	Good	Excel.	Excel.	Excel.	Poor to fa
Molding characteristics	Fair to excel.	Good	Good	Fair	Good	Very good

Engineering Design With

Description and properties of 15 important elastic polymers.

BY VARYING BASIC COMPOSITIONS, ingredients and processing of the five basic elastomers a wide range of rubber-like materials can be produced. However, a small but practical range is sufficient for such engineering applications as compression pads, impact bumpers, vibration insulators, engine and motor supports, dampers, suspension bushings, oil and grease seals, grommets and weatherseals. This group includes fifteen different compounds, classified under three different Shore Durometer hardnesses (40 Durometer is a soft grade, 50 Durometer is a medium hardness, and 60 Durometer is a firm hardness) of five different basic elastomers. Table I lists general properties of these elastomers, plus Thiokol (polysulfide "rubber").

Table II gives the conventional physical properties of these polymers and copolymers, as well as the SAE-ASTM grade numbers of the five elastomers compounded, in the three different Shore Durometer hardnesses. The key to the SAE-ASTM grade number system is given in Table III. The type of classification simplifies the writing of specifications for a wide range of compounds and conditions. As shown in Table II, there are three elastomers in the Type R non-oil-resistant classification, and two elastomers in the Type S oil-resistant

Table II—Physical Properties of Five Elastomers

Elastomer	Natural Rubber	Neoprene	Butadiene Styrene	Butadiene Acrylonitrile**	Butyl
40 Shore Durometer Group					
SAE-ASTM grade number	RN-430-ABF	SC-425-ABEF	RS-415-ABF	SB-410-ABEF	RS-420-AB
Ultimate tensile at 70 F, psi	3075	2800	1575	1325	2400
Elongation at rupture at 70 F, %	690	880	800	460	660
Shore Durometer hardness at 70 F	41	41	40	39	40
Comp. set, *"A" (constant load)	15.1	14.6	12.5	9.0	12.7
Comp. set *"B" (constant defl.)	22.6	22.1	18.5	27.4	14.8
Specific gravity	1.05	1.33	1.09	1.13	1.04
Vulcanization shrinkage, %	2.44	1.99	2.30	2.45	1.85
50 Shore Durometer Group					
SAE-ASTM grade number	RN-530-ABF	SC-520-ABEF	RS-518-ABF	SB-520-ABEF	RS-525-AB
Ultimate tensile at 70 F, psi	3650	2175	2050	2200	2975
Elongation at rupture at 70 F, %	640	740	710	510	710
Shore Durometer hardness at 70 F	52	49	50	49	51
Comp. set, *"A" (constant load)	12.0	20.2	14.6	7.2	13.5
Comp. set *"B" (constant defl.)	19.8	30.6	16.9	26.9	18.0
Specific gravity	1.11	1.37	1.10	1.21	1.085
Vulcanization shrinkage, %	2.16	1.61	2.13	1.91	1.66
60 Shore Durometer Group					
SAE-ASTM grade number	RN-630-ABF	SC-630-ABEF	RS-625-ABF	SB-617-ABEF	RS-625-AB
Ultimate tensile at 70 F, psi	3450	3275	2850	2250	2725
Elongation at rupture at 70 F, %	610	690	720	410	610
Shore Durometer hardness at 70 F	61	60	59	59	58
Comp. set, *"A" (constant load)	10.1	11.2	13.7	21.8	14.2
Comp. set *"B" (constant defl.)	20.0	24.6	20.0	24.0	19.3
Specific gravity	1.21	1.34	1.18	1.24	1.19
Vulcanization shrinkage, %	2.01	1.72	1.91	1.71	1.04

*Compression set is the residual decrease in thickness, expressed as a percentage of the original thickness, of a test specimen measured 30 min after removal from a loading device in which it had been subjected for a definite time to compressive deformation under specified conditions of load application and temperature. (See ASTM D395-47T.)

**The butadiene acrylonitrile compounds used had exceptionally high resistance to oils of high aniline point, and aromatic fuels.

Natural and Synthetic Rubbers

ELLWOOD F. RIESING
Manager, Automotive Engineering and Sales*
Firestone Industrial Products Division
Firestone Tire and Rubber Company

classification. These are all solid compounds.

Space does not permit presenting properties of sponge rubbers referred to in the classification chart, nor has any attempt been made to tabulate properties of either the large group of rubber-like flexible plastics (such as the vinyls), or the Type SA group of synthetics such as Thiokol.

The design of load-carrying rubber parts for vibration insulation, noise suppression, and shock impact insulation involves one or the other of two distinct types of stresses—in shear, or in compression. There are also designs that stress the elastomer in both shear and compression; these will be discussed later.

SHEAR DESIGNS. Designs requiring large deflections and low natural frequency oscillations cannot be adequately satisfied by stressing the elastomers in compression alone. Such designs should utilize the resiliency of the material by stressing it in simple shear when bonded to metal plates. Rubber is most compliant when used in simple shear, since the modulus of elasticity in shear when bonded to plates is approximately one-sixth the compression modulus.

Shear designs lend themselves to

* Now Manager, Sales Engineering, Industrial Division, National Motor Bearing Co., Inc.

Table III—SAE-ASTM Classifications and Grade Numbers

Classification*

Type R (non-oil-resistant)	Type S (oil-resistant synthetic)
RN–Rubber or reclaim	SA–Maximum oil resistance
RS–Synthetic rubber	SB–Good oil resistance
	SC–Medium oil resistance

Grades

Solid Rubber				Sponge Rubber			
Grade	Durometer	Grade	Tensile		Grade	psi***	
3	30± 5	00	Not req'd	Chemically	11	3½ ±	1½
4	40± 5	02	200 psi	blown	12	7 ±	2
5	50± 5	03	300 psi		13	11 ±	2
6	60± 5	04	400 psi		14	15 ±	2
7	70± 5	05	500 psi		15	20½ ±	3½
8	80± 5	06	600 psi				
9	90± 5	07	700 psi	Latex	21	0.19 ±	0.09
		08	800 psi	foam,	22	0.35 ±	0.09
		09	900 psi	cored	23	0.57 ±	0.13
		10	1000 psi		24	0.85 ±	0.15
		14	1400 psi		25	1.17 ±	0.17
		15	1500 psi		26	1.53 ±	0.19
		15-6**	1500 psi		27	1.93 ±	0.21
		17	1700 psi		28	2.37 ±	0.23
		18	1800 psi				
		20	2000 psi	Latex	31	0.50 ±	0.30
		25	2500 psi	foam,	32	0.80 ±	0.40
		30	3000 psi	uncored	33	1.45 ±	0.75
		35	3500 psi		34	2.50 ±	1.10
					35	3.50 ±	1.30
					36	4.60 ±	1.40
				Closed	41	2 to 5	
				cell	42	5 to 9	
					43	9 to 13	

Suffix Letters

A—Oven-aging requirement	J—Superior abrasion resistance
B—Low compression set requirement	K—Adhesion requirement
C—Weather-resistance requirement	L—Maxl resistance to moisture absorption
D—Load-deflection requirement	M—Max. inflammability resistance
E—Oil-aging requirement	N—Max. resistance to repeated impact
F—Flexible at − 40 F	P—Min. staining effect of enamel and lacquer
FF—Flexible at − 70 F	Z—An additional laboratory test over and
G—Max. resistance to tear	above those designated by the suffix letter
H—Max. resistance to severe flexing	listed above, or life or performance requirements

* Classification of the compound is determined by the predominating basic elastomer therein.
** A suffix number following the grade designates a special elongation figure.
*** Lb. per sq. in. required to compress original thickness 25%, based on 50 sq in. Scott indenter foot

Table IV—Shear Modulus at 30 percent Deflection (psi at 70 F—see Fig. 5)

Temperature, F	Natural Rubber	Neoprene	Butadiene Styrene	Butadiene Acrylonitrile	Butyl
40 Durometer Group					
	RN-430-ABF	SC-425-ABEF	RS-415-ABF	SB-410-ABEF	RS-420-AB
158	58	64	42	30	58
70	59	65	49	37	59
0	61	69	65	100	60
50 Durometer Group					
	RN-530-ABF	SC-520-ABEF	RS-518-ABF	SB-520-ABEF	RS-525-AB
158	82	97	80	44	70
70	86	107	104	71	71
0	90	145	139	183	78
60 Durometer Group					
	RN-630-ABF	SC-630-ABEF	RS-625-ABF	SB-617-ABEF	RS-625-AB
158	132	120	98	69	86
70	134	127	126	110	90
0	147	153	183	257	93

Table V—Compression Modulus Up to 20 percent Compression (psi, 70 F)

Temperature, F	Natural Rubber	Neoprene	Butadiene Styrene	Butadiene Acrylonitrile	Butyl
40 Durometer Group					
	RN-430-ABF	SC-425-ABEF	RS-415-ABF	SB-410-ABEF	RS-420-AB
158	400	470	360	212	340
70	405	475	415	270	360
0	410	480	810	2100	420
50 Durometer Group					
	RN-530-ABF	SC-520-ABEF	RS-518-ABF	SB-520-ABEF	RS-525-AB
158	540	690	480	340	410
70	635	700	620	500	460
0	690	900	880	3250	660
60 Durometer Group					
	RN-630-ABF	SC-630-ABEF	RS-625-ABF	SB-617-ABEF	RS-625-AB
158	990	900	660	460	540
70	1050	915	800	685	620
0	1250	940	1350	4300	900

simple, straight-forward solutions. The linearity of the stress-strain curves in shear for the elastomers is responsible for this. Fig. 1 is a typical shear modulus curve, clearly showing the very small variation in modulus for shear strains up to 30 percent deformation; the modulus falls off slightly and then picks up again as elongation of the molecular structure (tensile strain) makes itself evident. Compression modulus, however, does not vary linearly.

Hooke's law is partially obeyed by rubber-like materials stressed in shear. However, they follow it to a much lesser degree when stressed in pure tension, and do not follow it at all when stressed in compression. Under compressive stresses the stress-strain curve is definitely non-linear beyond 18 to 20 percent strain.

The basic form for a shear design is a *pair* of single shear sandwiches, as shown in Fig. 2. If greater deflections are necessary, the designer can resort to a double pair or quadruple pair. The normal working load should not strain the shear sandwich beyond 30 percent. For a *pair* of single shear sandwiches, following formulas apply:

$$d = \frac{WT}{AG_s}, \text{ and}$$

$$\text{shear rate per inch} = \frac{W}{d} = \frac{AG_s}{T},$$

where d = shear deflection, in.; W = load, lb; T = free molded thickness, in. (not laterally compressed thickness); A = twice the area at smallest cross section, sq in.; and G_s = shear modulus, psi (see Table IV).

Correction must be made in the shear modulus value where the ratio d/T is more than 0.3. The value of G_s must be multiplied by 0.9 where d/T is 0.35 to 0.8, and it must be multiplied by 0.833 if the value of d/T is 0.8 to 1.2. Properly designed shear sandwich elements should never allow movements beyond a d/T ratio of 1.0. As shown in Fig. 3, a compression buildup safety limiting device should be incorporated in the design to accomplish this.

Where square, rectangular, or other shaped shear sandwich constructions are used, the shear stress limit should be 20 to 30 psi, based on the smallest bonded area. Shock load stresses as high as 50 psi may be allowed. This applies to engine mountings, motor supports, resilient wheels, and rubber springs of various types, where set in the rubber is an important consideration. In such constructions, stress relief should be worked into the design by incorporating fillets in the rubber or radii on the edge of the metal part or both, since most fatigue failures

start at the edge of the bond. Fig. 4 shows a number of design principles for shear as well as compression.

The designer frequently resorts to lateral compression of the rubber in a shear sandwich, to facilitate anchoring of the metal elements. This changes the shear characteristics of the design only slightly for lateral compression loadings ranging from 10 to 50 psi, with shear calculations based on the original free thickness between the bonded plates.

Table IV gives moduli of elasticity in shear for deflections up to 30 percent, when load is applied parallel to the bonded surfaces, for the 15 compounds listed in Table II. The effect of temperature is shown in influencing the shear modulus. These shear modulus values, established by the author, were arrived at by considering the static and dynamic shear modulus values for vibration frequencies up to 240 cycles per minute. For designs subjected to frequencies above 240 cpm values for the specific frequency and temperature should be obtained from a rubber manufacturer having modern laboratory facilities. The difference between static and dynamic modulus at high frequency oscillations and at elevated or depressed temperatures may vary up to three times the static modulus.

Shear sandwich designs are practically free of "shape factor" influence —the ratio of width to length does not upset the design calculations. For example, if the adhesion and cross sectional areas and thickness are kept constant, the length may be equal to six times or only $\frac{1}{16}$ the width. Designs should be kept symmetrical whenever possible.

COMPRESSION DESIGNS. Although simpler in appearance than shear sandwich designs, compression designs are actually much more involved from an engineering standpoint, because they are completely dependent on the shape of the vulcanized elastomer. However, compression designs are used in the majority of resilient insulating elements, for vibration control, protection against impact shock, and for reducing the transmission of noise.

Most desirable results and longest life are obtained by bonding the rubber to plates, thereby preventing slippage and destructive abrasion of the load-carrying surfaces.

Designs employing rubber in compression should be so calculated to limit the deflection under most working loads to 10-20 percent of the free unloaded height of rubber thickness. For practical purposes, all resilient elastomers can be considered to have

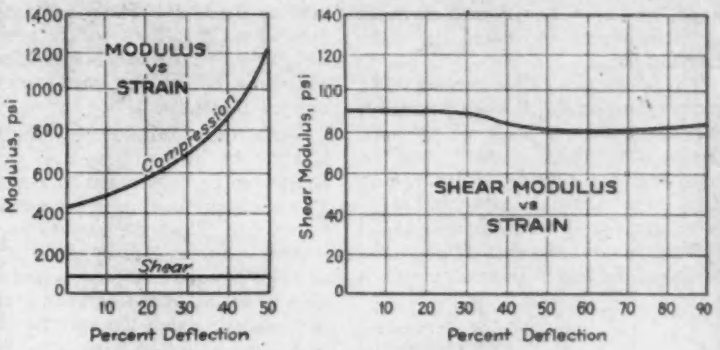
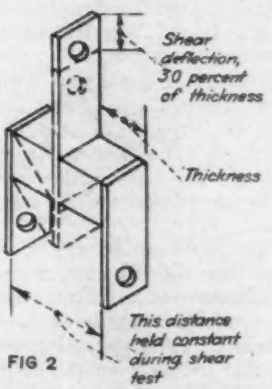


Fig. 1 — Typical shear modulus curve, compared with (left) compression modulus curve.



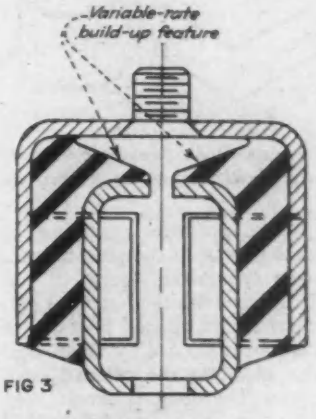


Fig. 2—Basic form for a shear design is a pair of single shear sandwiches. Fig. 3—To prevent excessive movement in shear sandwich elements, a compression build-up limiting device should be incorporated in design.

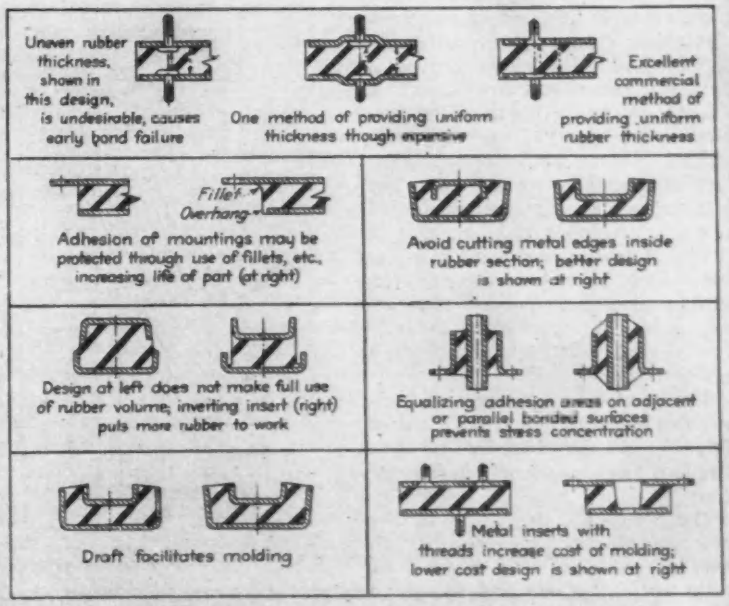

Uneven rubber thickness, shown in this design, is undesirable, causes early bond failure
One method of providing uniform thickness though expensive
Excellent commercial method of providing uniform rubber thickness
Fillet
Overhang
Adhesion of mountings may be protected through use of fillets, etc., increasing life of part (at right)
Avoid cutting metal edges inside rubber section; better design is shown at right
Design at left does not make full use of rubber volume; inverting insert (right) puts more rubber to work
Equalizing adhesion areas on adjacent or parallel bonded surfaces prevents stress concentration
Draft facilitates molding
Metal inserts with threads increase cost of molding; lower cost design is shown at right

Fig. 4—Principles for compression and shear design in rubber.

no volume compressibility. To obtain resilience the material must not be confined, but must be allowed to bulge at the surfaces normal to the load-carrying area.

Load-deflection characteristics of compression sandwiches are completely dependent on the shape of the final design. A "shape factor" must be considered for every set of dimensions. This shape factor, described in Fig. 5, is the quotient of one load area divided by the free side areas that are permitted to bulge. In Fig. 5 this bulge area is

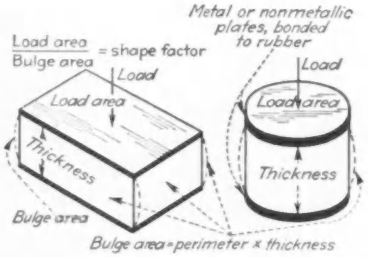


Fig. 5—Shape factor must be considered for every particular set of dimensions.

equal to the perimeter multiplied by the free thickness. If compression pads are made cylindrical, square or rectangular voids, with the long axis perpendicular to the bonded plates, then the bulge areas formed by these voids must be added to the external perimeter area. The designer should keep the dimensions of a compression pad or spring under consideration within shape factors of 0.2 to 1.2.

Compression modulus values vary with shape factor and percent of deflection. Since the stress-strain curve is not a straight line, compression modulus values change continuously with applied load. The author arrived at the compression modulus values in Table V by considering the static compression modulus at 20 percent strain and the dynamic compression modulus for frequencies up to 240 cpm. The butadiene-acrylonitrile compounds used here were developed for high resistance to oils of low aniline point and aromatic fuels. For load carrying elements subject to frequencies considerably above 240 cpm, actual modulus figures should be obtained from a rubber mounting.

The effect of temperature on compression modulus should be noted in Table V. As shown in Fig. 1, compression modulus is a function of strain.

The nomograph charts, Fig. 6, take into consideration the change in compression modulus with percent of strain (or deflection) and the shape factor, showing its limitations for design purposes. The author's data in Fig. 6 apply to natural rubber and all the synthetic rubber compounds covered in Table II. These nomographs can be used as follows, for compression designs where the free bulge sides are perpendicular to the load areas or bonded plates.

With the load to be supported known, the load-carrying area of the design can be determined by assuming a compression loading in the range 50 to 200 psi. Thus, the load to be supported, in lb, divided by the compression loading, in psi, equals the load area in sq in. A thickness must then be assumed. From the selected thickness, and with the load area determined, the shape factor can readily be established (see Fig. 5). The intersection of the shape factor, and the assumed compression loading in Fig. 6, establishes the compression modulus value at 10 and 20 percent strain of the compound required.

It should be noted that as the perimeter area of the pad keeps increasing (due to greater thickness or cored out openings), providing a larger bulge area, the necessary modulus increases if the same deflection is to be maintained.

The charts can be employed in reverse also; with the compression modulus known for a definite compound, the desirable compression loading in psi can be established. Interpolations can also be made on the charts. Results should be accurate within 10 to 15 percent.

The data in Fig. 6 resulted from a study of a large number of rubber compression pads and springs vulcanized of a variety of compounds of different elastomers, in various sizes, thicknesses, and shapes, in solid and hollow forms bonded to plates. A few examples of the wide variety of shapes which were investigated are shown in Fig. 7. These tests proved that similar compression elements bonded to metal plates, made of the identical compound and vulcanized to the same degree of cure, have the same stress-strain curve.

For compression sandwiches not having the free sides normal to the load carrying plates, the method of arriving at the shape factor must be modified.

SANDWICHES WITH SIDES OTHER THAN NORMAL TO BONDED SURFACES. The shape factor nomographs previously discussed are applicable to sandwiches of this type with angles up to 30 deg, if the shape factor computation is based on the ratio of the actual bonded area of one loaded surface to the bulge area, and if the loading is calculated on the basis of a simple arithmetic average between the actual bonded area of one loaded surface and the "projected area". The length L', Fig. 8, illustrates this projected area.

Parts of various areas, of constant thickness, were tested in compression with the angle θ varying from 0 to 60 deg. At the extreme angle of 60 deg the bonded plates no longer remain parallel, since the loading is uneven across the top and bottom surfaces, varying from a maximum at the small

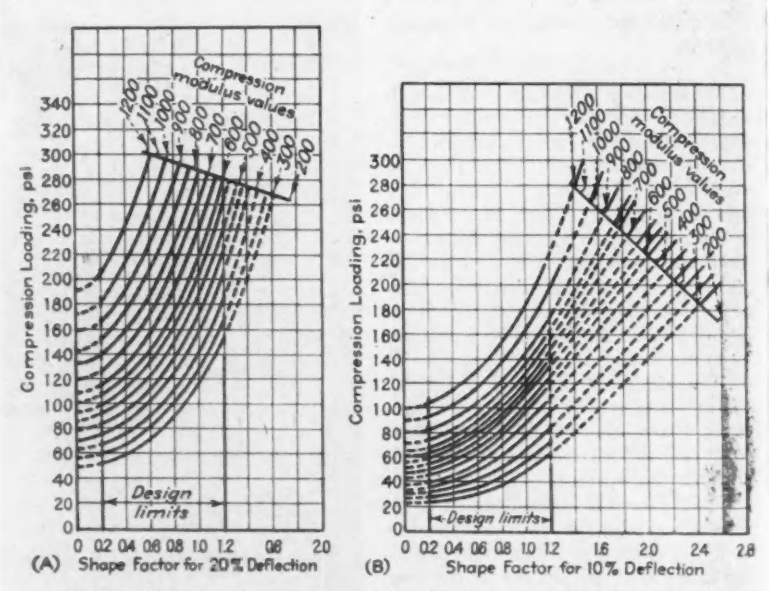


Fig. 6—Nomographs showing relationship between shape factor, compression modulus, and compression loading for elements made of natural rubber.

remaining projected area in the middle, to zero at the opposite extremeties. The plates of the sandwich tend to pull away from the platens of the machine.

The increase in bulge area of angular pieces, over sandwiches with sides normal to the load area, is controlled by the cosine of angle θ. Fig. 9(A) reveals that, as a result, the degree of loading required to produce a given deflection is a linear function of the cosine of this angle. These straight lines are contrasted with the curves in Fig. 9(B), in which the same load values are plotted against the angle θ in degrees. Although points for these two sets of curves were plotted for angles through 45 deg, a limit of 30 deg is suggested for best results.

The behavior of parts having cross sections which are symmetrical trapezoids, was shown to have no correlation with parts having a parallelogram cross section. However, a reasonable approximation can be obtained with the shape factor nomographs, if the shape factor is calculated on the basis of the arithmetic average of the two loaded surfaces and (since the smaller of these is also the projected area), the loading is figured on the basis of the smaller of the two loaded areas. This approximation is reasonably accurate, however, only for parts having an angle θ up to 25 degrees.

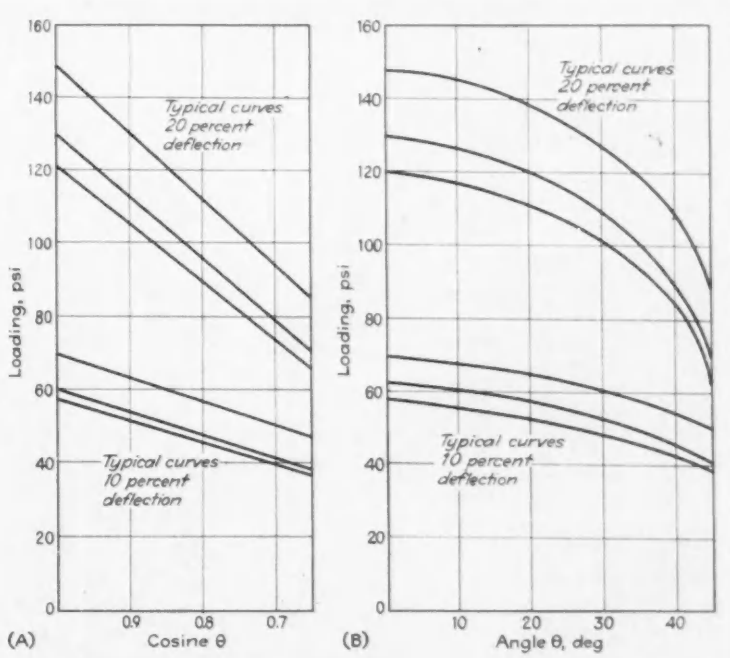


Fig. 9—(A) Increase in bulge area of angular pieces varies with loading. (B) Plotting directly against angular gives curves, in contrast to straight lines produced to plotting against cosine of angle (A), which is actual area difference caused by change in shape.

BUMPERS. Bumper compression pads subjected to periodic impact warrant investigation by the designer of impact resilience values, which can be determined by several methods. The ball rebound device is an easy way to compare various compounded elastomers and is an inexpensive inspection method on molded shapes. The ball rebound device permits calculation of a factor which can be called impact resilience. This value is expressed by dividing the first rebound height H_1 by the distance through which the ball is originally dropped, H_0.

$(H_1/H_0) \times 100$ = impact resilience, %.

EDITOR'S NOTE: A second and concluding article by Mr. Riesing, covering general design, will be presented in the December PRODUCT ENGINEERING.

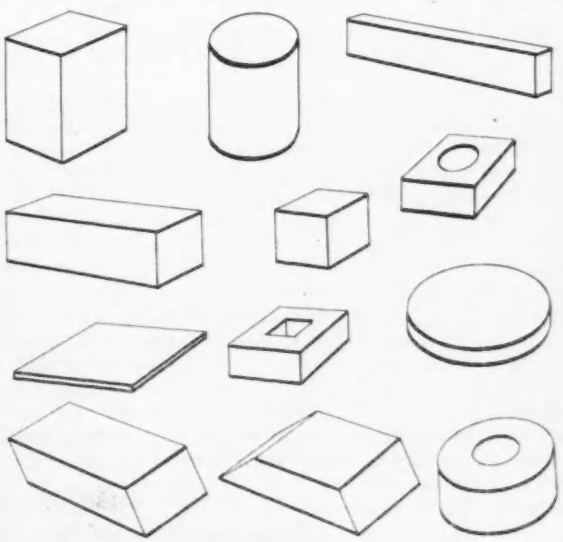


Fig. 7—Typical shapes investigated, in a variety of compounds, to establish nomographs in Fig. 6.

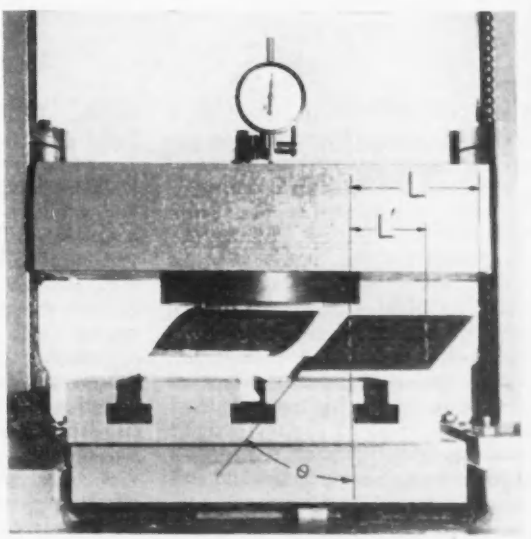
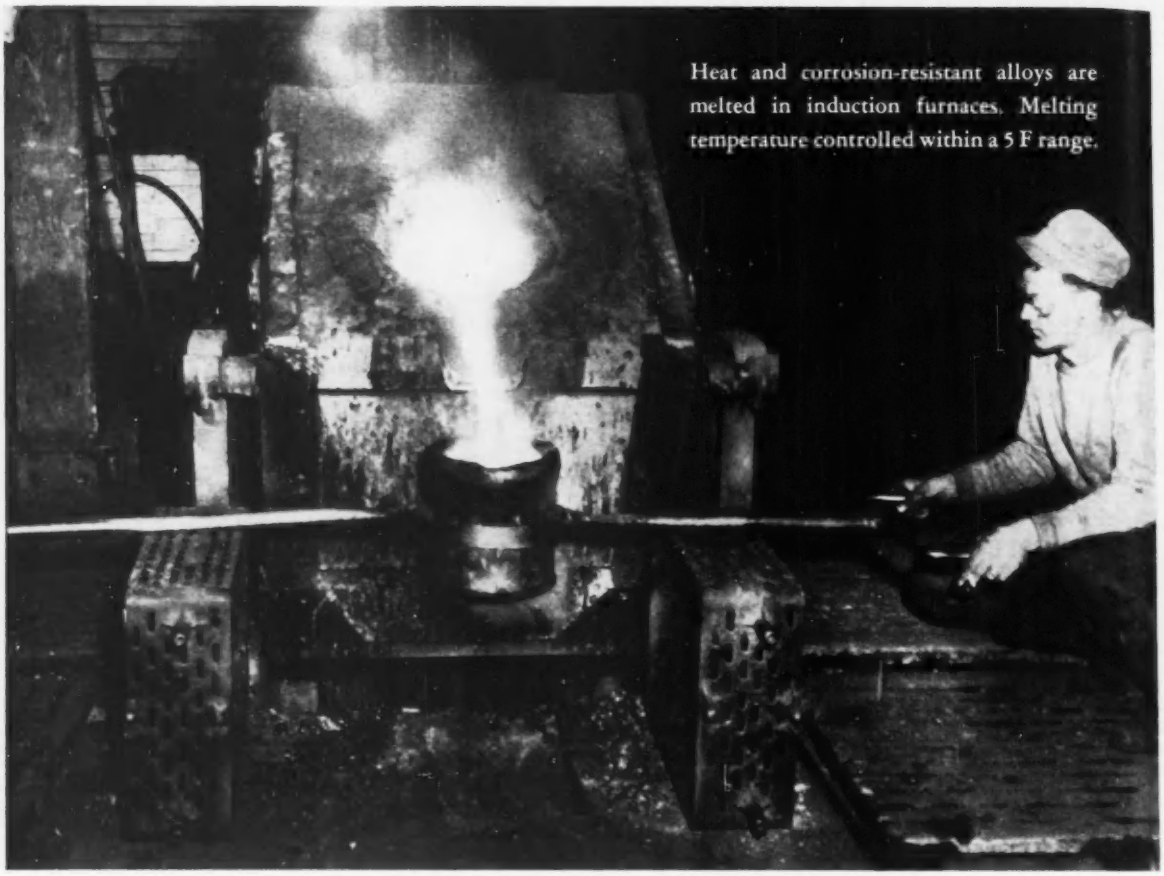


Fig. 8—Typical sandwich with sides other than normal to bonded surfaces. Length L' is projected area.



Heat and corrosion-resistant alloys are melted in induction furnaces. Melting temperature controlled within a 5 F range.

Centrifugal Casting in

DEVELOPMENT OF MODERN TECHNIQUES for centrifugal casting in permanent molds has enabled improved and expanded production of cylindrical shapes, in a variety of alloy steels. Although pipe tubing, bushing and gear blanks had been centrifugally cast in sand molds, the application of metal molds to highly alloyed centrifugal castings is comparatively new. It is a result of the need for circular cast shapes capable of meeting X-ray inspection and other high standards of acceptance, required for jet engine parts and other precision components subjected to high temperature and corrosive conditions.

The Process

A variation of "true" centrifugal casting, the new process utilizes permenent molds instead of refractory molds. In this method, the circular mold is rotated about its axis, either horizontal or vertical, while the molten metal is poured and the casting cools and solidifies. No center core is used, since centrifugal force causes the liquified metal to assume the shape of the mold into which it is poured. Cooling is from the outer surface toward the center.

Permanent metal molds used in this process are made from specially developed steels, with choice of die material depending on casting shape, material and melting temperature. Metal dies offer several advantages over refractory materials: Mold erosion is eliminated; casting grain size is generally smaller; and production costs are lowered through long life of the mold.

After the ring-shaped dies are mounted and conditioned by preheating and spraying, the metal to be cast is melted in an induction furnace. Since the casting bore diameter depends on the amount of molten metal poured into the die, the weight of the metal charge is closely controlled, to produce a ring of proper dimensions. The molten metal is poured into the rotating die of the spinning machine, with speed of rotation varying from 100 to 1,200 rpm, depending on the design and size of the casting. The poured casting is spun until solidification is complete.

Advantages

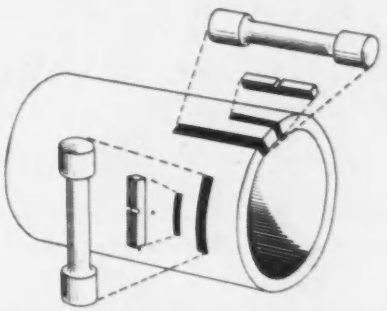
Permanent mold centrifugal casting is claimed to have the following advantages, when compared with conventional methods:

1. Higher quality castings, result-

Table I—Physical Properties of Centrifugally Cast Heat-Resistant Steels

12% Cr Type: {Composition—C, 0.10%; Cr, 12.50%
{Heat Treatment—Normalized and Tempered.

Property	Longitudinal Specimen	Tangential Specimen
Tensile Strength, psi	101,000	108,000
Yield Strength, 0.2% offset, psi	83,000	87,500
Elongation in 2 in., %	23.0	24.5
Reduction of Area, %	63.0	60.5
Izod Impact, ft-lb	45—52	.....
Charpy Keyhole Impact, ft-lb	25, 26, 26	26, 25, 25
Brinell Hardness No.	223—228	228—235
Endurance Limit, psi	55,000	.....



How longitudinal and tangential test bars and Izod impact specimens are cut from centrifugal casting.

19-9-Cb Type: {Composition—C, 0.07; Cr, 19.0; Ni, 10.0; Cb, 0.90
{Heat Treatment—Solution, Air Cooled.

Property	Longitudinal Specimen	Tangential Specimen
Tensile Strength, psi	79,000	78,000
Yield Strength, 0.2% offset, psi	40,500	39,500
Elongation in 2 in., %	47.0	46.0
Brinell Hardness No.	156	156
Charpy Keyhole Impact, ft-lb	26, 25, 25	25, 24, 25

19-9-Mo Type: {Composition—C, 0.07; Cr, 19.0; Ni, 9.5; Mo, 2.5
{Heat Treatment—Solution, Water Quenched.

Property	Longitudinal Specimen	Tangential Specimen
Tensile Strength, psi	76,500	78,000
Yield Strength, 0.2% offset, psi	40,500	43,500
Elongation in 2 in., %	60.0	55.5
Brinell Hardness No.	156	156
Charpy Keyhole Impact, ft-lb	70, 67, 68	66, 63, 65

Permanent Molds

WARDEN F. WILSON
Lebanon Steel Foundry

Wider use of high alloy steels for heat- and corrosion-resistant applications is limited only by available fabricating methods. Centrifugal casting in permanent molds is a relatively high-volume, low cost method of producing symmetrical shapes in such materials, with great accuracy, strength, and internal soundness.

ing in enhanced physical properties, including greater density.

2. Production of complex structural shapes which cannot be satisfactorily cast by static methods.

3. Uniform strength throughout the piece; no flow lines, as in forgings.

4. Difficult or impossible-to-forge alloys, used because of extreme heat-resistance and high hardness required, can be centrifugally cast.

5. Stainless steel compositions can be used which are up to about 30 percent more machinable than wrought, forged or seamless tubing materials.

Density and soundness of the product is unusually high, because the centrifugal force pushes the metal against the walls of the metal mold, and brings any minor impurities to the internal surface, which is subsequently machined off. Solidification takes place progressively and no center-line shrinkage occurs. The molten metal freezes with a normal pattern, which is extremely fine on the outer surface, because of the rapid rate of chilling and coarsens somewhat toward the inner surface.

Some of the physical properties of longitudinal and tangential 12 percent chromium (AISI Type 410) steel specimens taken from centrifugal castings are shown in Table I. The accompanying drawing shows how specimens are cut out of a casting for

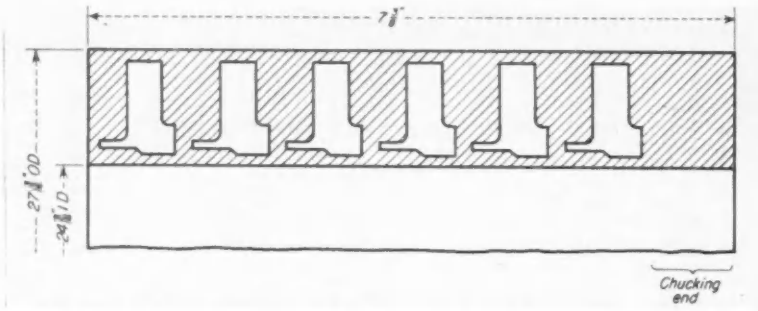


Fig. 1—Rings used in jet engines show economy of cutting multiple units from one blank.

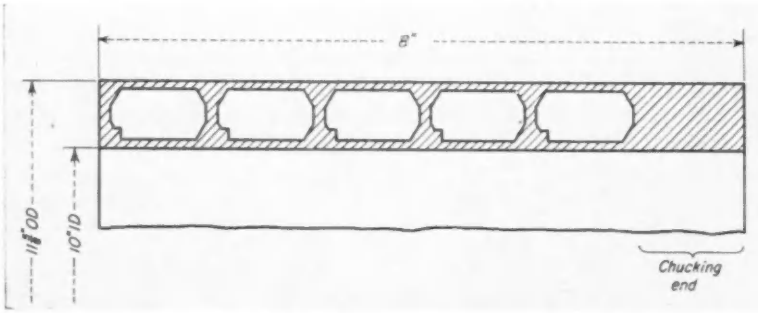


Fig. 2—Metal conservation and reduced machining time can be achieved through centrifugal casting method as shown by grouping of multiple rings in the production of high alloy valve seats for 10 in. gate valves, of the type used in the chemical industries.

Table II—Alloy Steels Suitable for Centrifugal Casting in Permanent Molds

(Heat and Corrosion Resistant Alloys)

Steel Casting Specifications		Nominal Analysis, percent					Engineering Application
Lebanon Desig.	Specification	C	Cr	Ni	Mo	Others	
(L) 12	ASTM A-296-46T Grade 10 AMS 5351 AISI 410	0.10	12.50	0.50			Heat-treatable
(L) 21	ASTM A-296-46T Grade 5 AMS 5363 AISI 347	0.07	19.50	10.50		Cb, 0.80	Weldable grade for high temperature applications
(L) 22	ASTM A-296-46T Grade 1 AISI 304	0.07	19.50	9.0			General corrosion resistant alloy
(L) 22XM	ASTM A-296-46T Grade 4 Type 316	0.07	19.50	10.00	2.25		Corrosion resistance and strength at moderately high temp.
(L) 30	ASTM A-296-48T Grade 7	0.15	24.50	13.50			Heat resistant alloy
(L) 34		0.07	20.50	28.50		Cu, Mo, Si	Highly resistant to sulfuric acid
(L) 46	ASTM A-296-46T AMS 5365	0.14	24.50	20.50			Heat and oxidation resistant
19-9-DL	AMS 5369	0.30	19.00	9.00	1.25	Cb, W, Ti	Heat resistant
(L) 130	H.R. Crown Max (Br.)	0.25	23.00	12.50		W, 3.00	Heat resistant
N-155	AMS 5767	0.15	21.00	20.00	3.00	Co, 20.00 W, Cb	Heat resistant
(L) 46	ASTM A296-49T Grade 8	0.20 max	25.00	20.50			Heat resistant

tensile and Izod impact tests. There is some variation in tensile and yield strength values obtained from tangential and longitudinal specimens. However, impact tests made in each of these directions show values consistent within 1 ft-lb. Izod specimens in both directions show values ranging from 45 to 52 ft-lb, which indicates exceptional impact resistance from high tensile materials.

The endurance limit of the Type 410 centrifugal casting itself for 10,000,000 cycles is 55,000 psi. Ductility values of this type of casting are exceptionally high, considering the tensile strength and the Brinell Hardness of the product. The Brinell numbers themselves are quite uniform over a long series of tests.

Widely used for centrifugally cast heat-resistant parts is Type 347 (19 percent chromium-9 percent nickel) containing columbium. This particular centrifugal casting shows high physical properties, with practically the same tensile strength in the longitudinal or tangential directions, as shown in Table I. Charpy keyhole impact specimens show values from 24 to 26 ft-lb, regardless of direction of specimen or position of the notch. The Brinell Hardness values are exceptionally uniform between the outside and inside diameters.

A third alloy steel adapted to centrifugal casting requirements is Type 316 containing molybdenum, corrosion-resistant alloy of the 19-9 series. Table I shows tensile test results on longitudinal and tangential specimens, which are high. Charpy impact keyhole specimens show this material to be uniform in properties throughout.

Another material currently used for jet engine parts is Type 310, a 25 percent chromium-20 percent nickel corrosion-resistant high temperature alloy, which has excellent physical properties after water quenching.

These alloys are widely used for gas turbine parts, in which high temperature properties are of utmost importance. A high degree of internal soundness is mandatory in such parts. Evidence of test data, together with actual service records of such parts, indicates centrifugal casting in permanent molds meets these rigid demands. The high alloys that can be produced are not limited to the four compositions already mentioned. Heat and corrosion resistant alloys, and carbon and low alloy grades, listed in Tables II and III, are equally adaptable to this process. Other grades, especially alloys that are difficult to forge, can be produced on special request.

Almost all alloys adaptable to sand

casting can be cast by centrifugal methods; improvement in mechanical properties is obtained with most metals. Tolerances typical in sand casting prevail in centrifugally cast parts. Some true centrifugal castings can be held to plus or minus 1/32 in. on outside diameters and in uniformity of wall thickness. Even finer tolerances are possible with careful foundry procedure.

Design Factors

Redesign of centrifugal castings calls for thoughtful consideration of process limitation, to obtain optimum benefits. The method is especially worthy of consideration for circular machinery parts, which can be cast to finished or substantially finished form.

In centrifugal casting it is preferable not to use more than one external flange. Although radii adjoining flanges and other abrupt changes of contour are not absolutely necessary in true centrifugal casting, use of a definite radius to avoid sharp corners will prolong mold life. Also, although the external shape of centrifugal castings can be almost any symmetrical shape, the importance of balancing molds for large parts should be kept in mind.

Variations in the cylindrical cross-section, such as flanged rings or bushings are achieved by machining the rough casting. It is common practice to machine a number of rings from each casting or cylinder blank, as shown in Fig. 1. Jet engine components produced economically in this manner include turbine shroud rings, turbine labyrinth seals, support rings, outer nozzle rings, nozzle guide-vane support rings, and turbine stator rings. Fig. 2 shows how producing multiple parts from a single blank achieves metal conservation and reduces machining time in the production of high alloy valve seats for 10 in. gate valves, by grouping multiple rings in one casting. Fig. 3 shows how uniformity of metal structure can be obtained in intricate cross-sections.

Some typical heat-resistant rings centrifugally cast in permanent metal molds are illustrated in Fig. 4. Many variations of diameter, thickness and length of the rings can be obtained. The rings shown have been machined to specified shape and size after casting. Facilities currently available can produce cylinders or blanks up to approximately 42 in. in diameter and up to approximately 3 ft in length, depending on the diameter. Castings are made in either vertical-or horizontal-axis machines, the choice being governed by the ratio of length to diameter. Fig. 5 indicates such ratios

Table III—Carbon and Low Alloy Steels Suitable for Centrifugal Casting in Permanent Molds

Steel Casting Specifications		Nominal Analysis, percent				Engineering Application
Lebanon Designation	Specification	C	Cr	Ni	Mo	
(L) B	NAVY B, 49-S-1 ASTM A-27-48 Grade 65-35 SAE 0030	0.25				General structural welding grade
(L) 3*	SAE 4140	0.40	1.00		0.25	High strength low alloy steel
(L) 205A*	NAVY F 49-S-1 ASTM A-148-48 Grade 90-60 SAE 8630	0.30	0.60	0.60	0.20	High strength low alloy heat-treatable structural steel
(L) 205B*	ASTM A-148-48 Grade 105-85	0.30	0.60	0.60	0.20	High strength structural alloys
(L) 205C*	ASTM A-148-48 Grade 120-100	0.30	0.60	0.60	0.20	Heat-treatable
(L) 205D*	ASTM A-148-48 Grade 150-125	0.30	0.60	0.60	0.20	Heat-treatable
(L) 8	ASTM A-217 Grade WC-9	0.15	2.40		1.00	High temperature alloy to 1,050 F. Good creep resistance
(L) 10	ASTM A-157-44 Grade C-5A AMS 5602	0.20	5.50		0.50	High temperature alloy to 1,100F

* These alloys are subject to normalizing, quenching, and tempering as required.

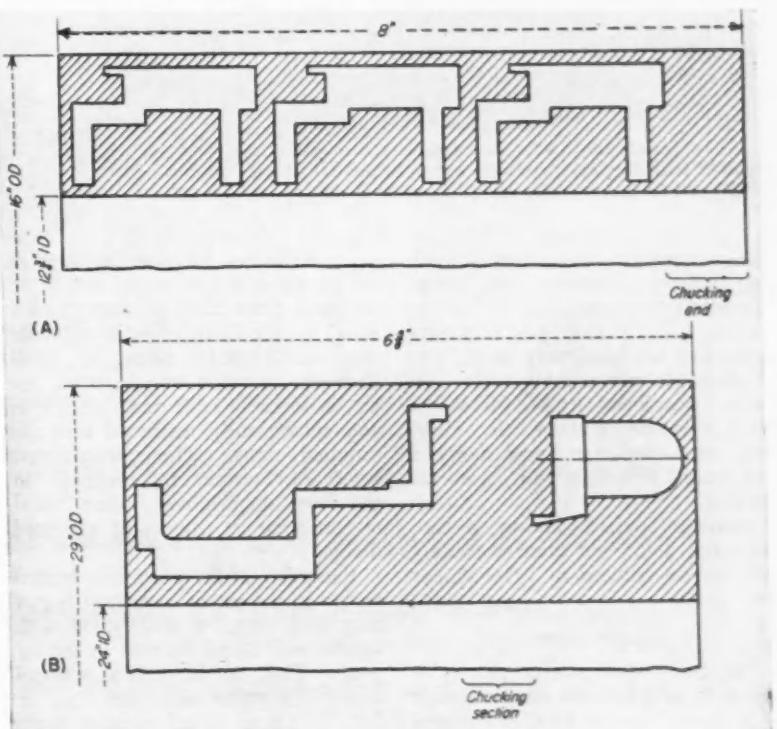


Fig. 3—Uniformity of structure can be obtained in intricate cross-sections, by careful grouping. Centrifugal casting in permanent molds produces higher quality alloy steel castings, resulting in enhanced physical properties and greater density.

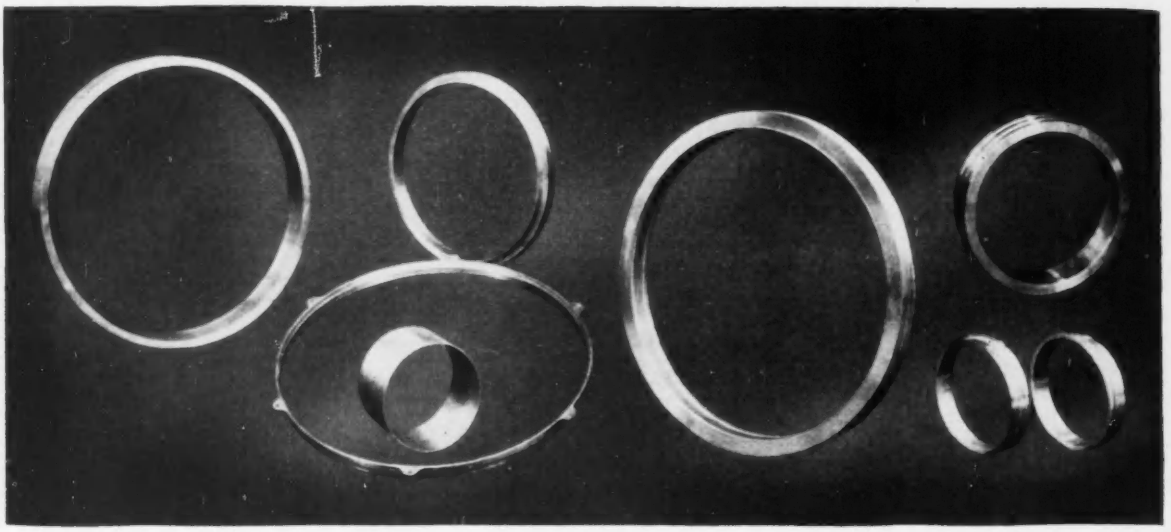


Fig. 4—Typical alloy steel rings centrifugally cast in permanent molds. This process enables a wide variety of shapes and sizes.

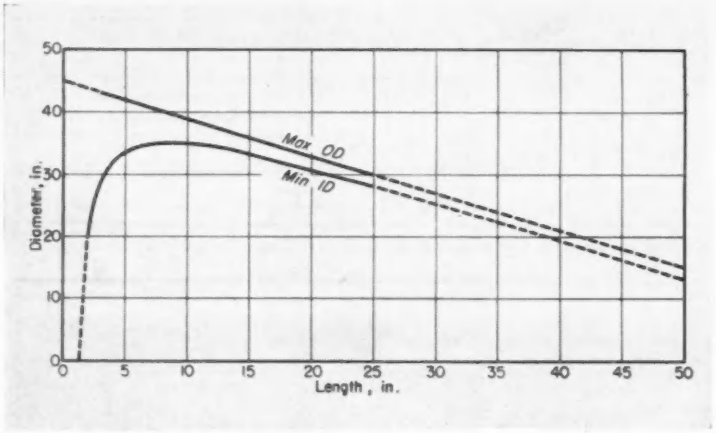


Fig. 5—Representative sizes that can be centrifugally cast from 1,000 lb of metal by available facilities (there are no theoretical limits). Other combinations of OD and ID are possible, depending on wall thickness and length.

Fig. 6—Deep-etched section of a centrifugal casting showing typical structure. Solidification is progressive from outer surface inward.

for parts produced by currently available facilities. However, there are no theoretical limitations on the size or weight of the cast parts; present limitations are based only on the size of machines and melting units. At present, maximum length for shapes up to 14 in. dia, is about 2 ft. However, new machines being designed will make 5 ft lengths with 14 in. dia possible.

Minimum wall thicknesses, as-cast, are about $\frac{3}{8}$ in., but after machining this can be reduced to 1/16 in., or less.

Applications

The centrifugal casting process is especially adaptable to alloys capable of withstanding the heat, centrifugal forces, thermal stresses, corrosion and vibration encountered in aircraft gas turbines. One austenitic, heat-resisting steel is cast in shapes as large as 30 in. dia and more than 500 lb in weight. From such cylinder blanks, either in the form of wide diameter rings or lengthier tubing of small diameter, separate rough rings are cut on specially built rotary cold saws and subsequently machined into the required rings. These components have been successfully employed in the Pratt & Whitney "Nene," and several other jet engines of advanced design.

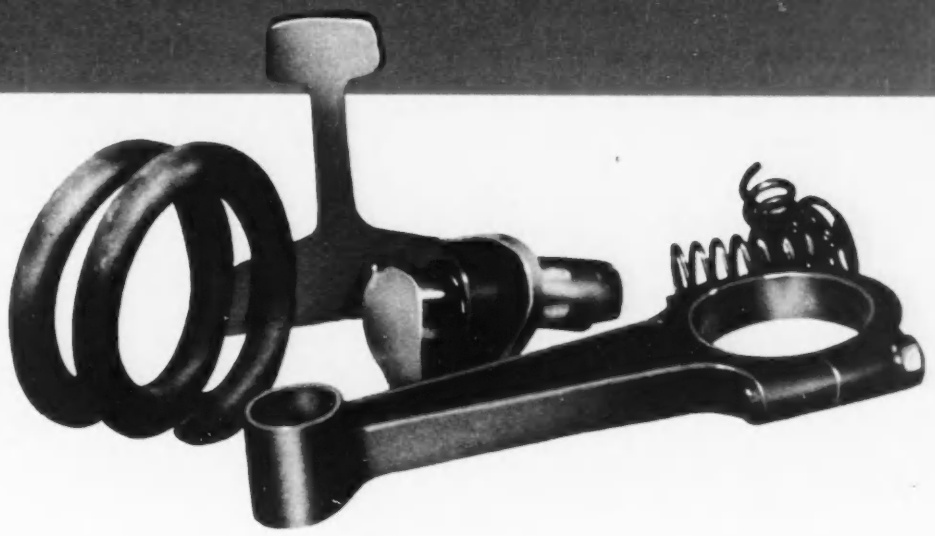
Versatility of the process is demonstrated in the use of centrifugally cast alloy steel parts in conjunction with welded and rolled sections in jet engines. They can be used as screwed inserts for valve seats, and can, in fact, be placed in any circular piece, whether forged or rolled.

Centrifugally cast pieces over 8 in. dia compare very favorably in cost with forged or rolled tubing: the larger the diameter, the more economical is the process; and in alloys that cannot be forged or rolled, the process is more economical for all sizes.

Aside from the jet-engine applications previously discussed, the process is readily adaptable to the production of a variety of bushings, valve seats, pump liners and other cylindrical and circular shapes in a wide range of alloys. It offers particularly significant possibilities for production of circular shapes required by the chemical, textile, oil, paper and pulp industries.

Experience in the production of aircraft engine components has shown this process to be a relatively cheap, and extremely reliable method of producing parts in special steels which, by virtue of their special properties, are difficult to fabricate by ordinary methods.

Fatigue WEAKNESS OF SURFACES



J. O. ALMEN

RESEARCH LABORATORIES DIVISION, GENERAL MOTORS CORPORATION

Shortcomings in current testing procedures and concepts that should be corrected by careful analysis and evaluation of the weakness of the surfaces of materials in fatigue. Machine parts are so far removed from ideal laboratory fatigue specimens that the latter are misleading as measures of worth.

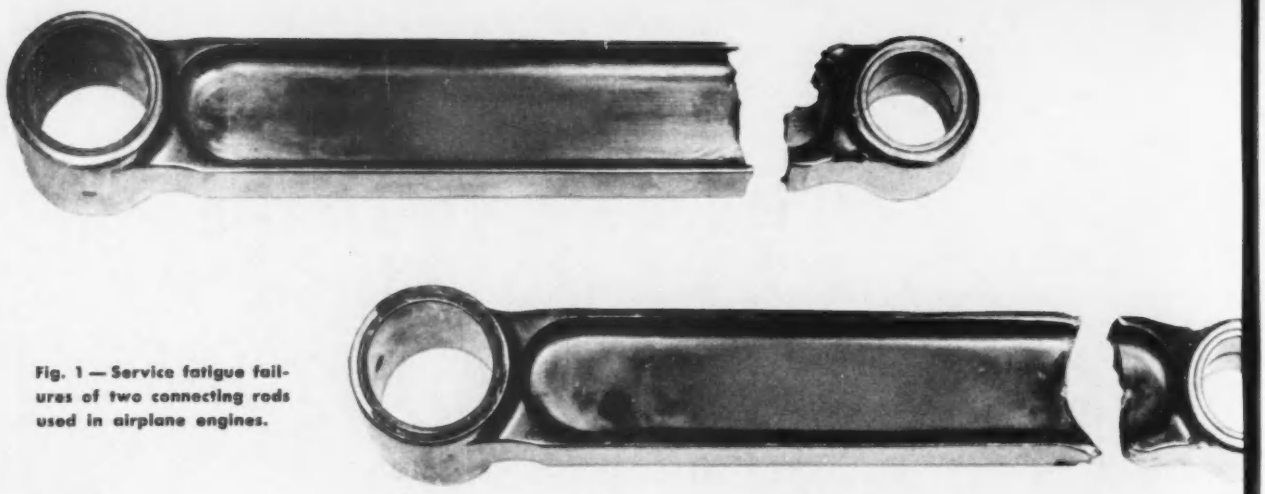


Fig. 1 — Service fatigue failures of two connecting rods used in airplane engines.

Strength Theories Are Misleading

THE commonly accepted theories by which we try to explain fractures that occur in structural materials are based upon data obtained from static tests. These theories work well enough when applied to the kind of loads from which they were derived, but they fail when the conditions of loading are changed. We are now slowly awakening to the fact that these theories are incomplete, inaccurate, and often misleading.

The remarkable progress that has been made toward more effective use of materials in modern machine structures has been accomplished in spite of our faulty understanding of the behavior of metals under the various kinds of loads that are encountered in practice.

Because of inadequate understanding of the fundamentals that are involved, designers of high duty machines cannot proceed by orderly application of engineering principles. Instead, machine parts are dimensioned by a curious mixture of engineering formulas, empirical correction factors, and plain guessing based on unorganized experience. Under the present circumstances, experience is far more important in solving machine design problems than training in engineering "fundamentals."

No doubt progress will be more rapid and superior designs will be produced as we more clearly comprehend the fundamental nature of failure under each of the various methods of loading. It is hoped that the following discussion of surface weakness will help in some small measure toward a more complete understanding of problems associated with strength of materials including the role, for good or for evil, that is played by manufacturing processes.

Fatigue Weakness of Surfaces

Surfaces of structural materials, regardless of smoothness, are much weaker in fatigue than are sub-surface materials. As indicated by available data, the surface strength of steel fatigue specimens may be only a half that of the sub-surface metal. Because of the great difference in these fatigue strengths, almost all the tests that are intended to establish the fatigue strength of metals succeed in measuring only the strength of the metal surface.

Forged surfaces, machining marks, sharp notches, corroded areas, abrupt changes of section, and other discontinuities have long been known to detract from the strength of fatigue specimens. Obvious surface imperfections are now recognized as stress raisers; and many fatigue tests have been made to measure the extent of the damage caused by each major form of surface imperfection. The fatigue strength of specimens having "imperfect" surfaces is compared to the fatigue strength of specimens that are so carefully processed that all visible stress raisers are removed. These carefully finished specimens are erroneously assumed to yield true measures of the fatigue strength of the metals being tested.

The approved methods for determining the inherent fatigue strength of structural materials fail in their purpose because:

1. The processes applied in finishing the specimens introduce residual stresses of significant but unknown magnitude.

2. Under dynamic loads the elastic limit of ordinary structural metals is so low (Ref. 1) that plastic deformations, the extent of which varies with the manner of loading, often alter the stress continuously during a fatigue test.

3. The surfaces of structural materials are weaker in fatigue (Ref. 2) than sub-surface material.

Shot Peening Theories

Shot peening induces residual tensile stress in the unpeened sub-surface metal as well as residual compressive stress in a thin surface layer. The residual tensile stress in the "core" increases as the depth of the peened layer is increased. Also for constant depth of peening, the tensile stress in the core increases as the thickness of the specimen is decreased. The internal tensile stress must vary with the depth of peening and with the specimen thickness since, to satisfy equilibrium conditions, the compressive force in a surface layer that results from peening must be balanced by an equal tensile force in the core metal.

It will be shown that the residual tensile stresses in the cores of cylindrical shot peened specimens of various diameters are approximately the same magnitude as the presumed difference in stress between the surface and the metal immediately below the peened layer. And also, that the magnitude of the residual stress induced in the surface by the peening will be compressive, Fig. 2, and approximates one-half of the nominal yield stress of the peened metal.

The increase in fatigue strength that follows shot peening is explained by two theories.

One theory holds that: (a) Surfaces are weaker under repeated loading than sub-surface material; and (b) Fatigue fractures can develop only from tensile stresses (Ref. 2). By this theory, the fatigue strength is increased because shot peening induces residual compressive stress in a thin surface layer (Ref. 3). As a result, the surface tensile stresses from external loads are reduced by an amount equal to the effective induced residual compressive stress. The observed gain in fatigue strength, therefore, results mainly from reducing the damaging tensile stress in the weaker surface metal with, perhaps, some increase in hardness.

The alternate theory assumes that cold working operations, such as shot peening, strengthen the surface layer by "work hardening" (Ref. 4). The resulting gain in fatigue strength, therefore, is a measure of the increase in strength that results from "work hardening," unless the fatigue fracture originates in the undisturbed metal immediately below the "work hardened" layer. The unpeened metal below this layer is assumed to withstand greater fatigue loads because, in beams subjected to bending loads, the stress in the sub-surface metal is less than the surface stress. The stress differences between the surface metal and the metal immediately below the peened layer must therefore be equal to or greater than the assumed gain in strength from "work hardening."

By the use of conventional assumptions, this stress difference can be calculated if the depth of the peened layer is known.

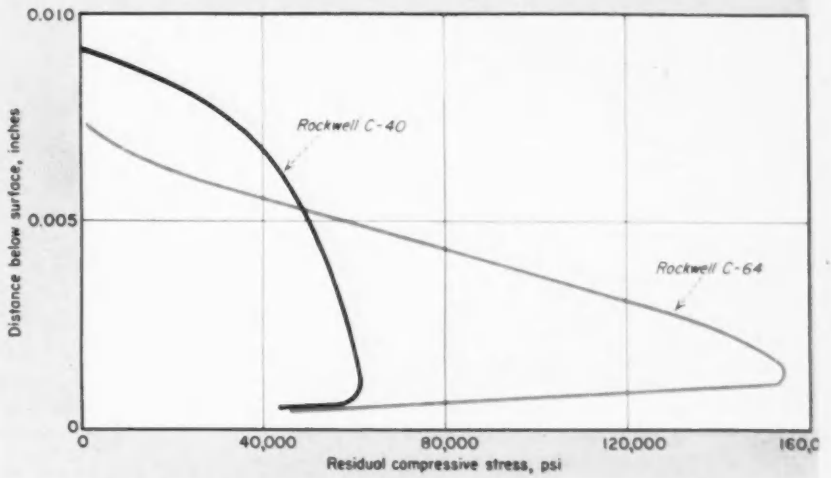


Fig. 2 — Magnitude and depth of the residual stress induced by shot peening varies with the yield stress of the peened metal.

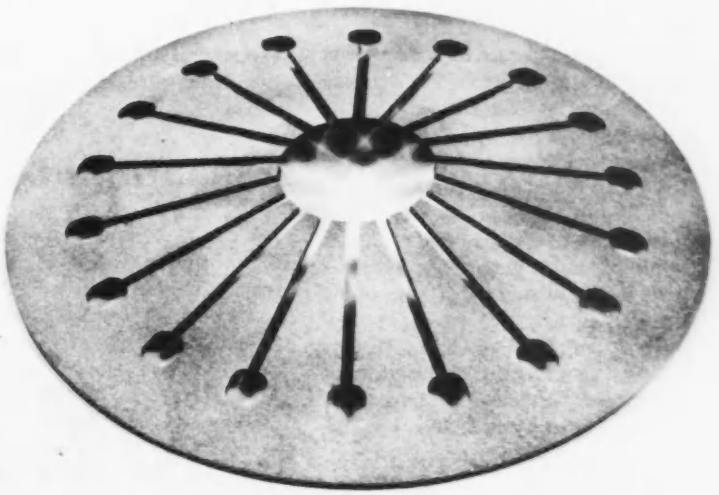


Fig. 3 — Shot peened Belleville spring. Life of this spring and of many other parts is increased by production shot peening.

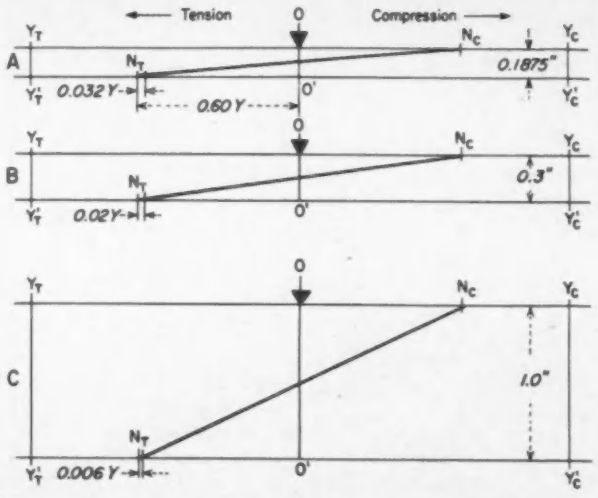


Fig. 4 — Stress gradients and stress reduction at 0.005 in. depth in beams of different thicknesses when subjected to bending loads equal to 60 percent of yield stress.

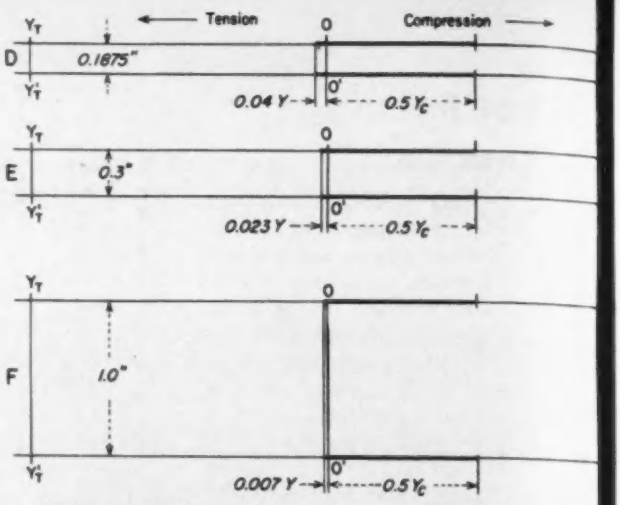


Fig. 5 — Residual compressive stresses in the surfaces and residual tensile stresses in the cores of different cylindrical specimens that have been shot peened 0.005 in. deep.

Stress Gradient in Beams

Within the elastic range, the stresses produced in beams externally loaded in bending are assumed to increase linearly from zero at their neutral planes to a maximum in the outermost fibers. The relative magnitudes of the stresses in the sub-surface metal and the stress at the surface, are proportional to the distances from the neutral plane.

In Fig. 4 the diagrams *A*, *B* and *C* show the stress gradients from bending loads in three cylindrical beams. The diameters of the beams are 0.1875; 0.3; and 1.0 in. respectively. The bending loads, which act on the beams in a downward direction, stress the outermost fibers to 60 percent of the nominal yield stress *Y* of the metal. At any depth in each beam, the horizontal distance from the vertical line *OO'* to the diagonal line N_cN_t is the resulting nominal stress. The nominal yield stress of the specimens is the horizontal distance from the vertical line *OO'* to the vertical line Y_tY_t' for tensile yield, and to the line Y_cY_c' for compression yield.

In the smallest beam, diagram *A*, the distance from the zero stress plane to the surface is one-half the beam diameter or 0.094 in. The nominal stress at a fiber 0.005 in. below the surface is

$$\frac{0.60\ Y\ (0.094-0.005)}{0.094} = 0.568\ Y$$

or 43.2 percent less than the nominal yield stress of the metal, and 3.2 percent of *Y* less than the nominal surface stress.

The nominal stress at a fiber 0.005 in. below the surface of the 0.3 in. specimen, diagram *B*, is 0.58 *Y*, or 42 percent less than the nominal yield stress of the metal, and 2 percent of *Y* less than the nominal surface stress. In the 1.0 in. beam, diagram *C*, the nominal stress at a fiber 0.005 in. below the surface is 0.594 *Y*, or 40.6 percent less than the metal nominal yield stress, and 0.6 percent of *Y* less than the nominal surface stress.

Equilibrium Conditions

The residual compressive stresses in the surfaces and the residual tensile stresses in the cores of shot peened cylindrical specimens that are required to satisfy equilibrium conditions are shown in Fig. 5. These specimens are of the same diameter as the bending fatigue specimens of Fig. 4, and assumed to be shot peened to a depth of 0.005 inch.

In the diagrams of Fig. 5, the nominal yield stress of the specimens is the horizontal distance from the vertical line *OO'* to the vertical line Y_tY_t' for tensile yield, and to line Y_cY_c' for compression yield. The distance $0.5Y_c$ is the maximum residual compressive stress induced by shot peening.

The resultant residual compressive force, which tends to lengthen the specimens, is exerted upon a hollow cylinder having an outer diameter equal to the diameter of the specimen and a radial thickness equal to the depth of peening.

Since the magnitude of the residual compressive stress from peening decreases with depth, the mean residual stress over the depth of the peened metal may be taken as two-thirds of the maximum residual stress induced by peening, or one-third of the nominal yield stress. From these values of the hollow cylinder area and mean stress acting upon that area, the force acting to lengthen the peened specimens may be calculated.

The resisting tensile force, which necessarily is equal to the lengthening force, may be assumed to act uniformly over the unpeened core area. From these data the magnitude of the resultant tensile stress in the core can be calculated.

By this procedure, the residual tensile stress in the unpeened core of the 0.187 in. dia specimen *D*, Fig. 5, is found to be approximately four percent of the nominal compressive yield stress. Similarly, the residual tensile stress in the cores of the specimens *E* and *F* are found to be 2.3 and 0.7 percent, respectively, of the nominal compressive yield stress.

Residual Core Stress Cancels Stress Gradient

When the peened specimens of Fig. 5 are subjected to the same bending loads as the specimens of Fig. 4, the maximum tensile stress in the core is the sum of the stress on a fiber 0.005 in. from the bottom surface of the beam, as calculated by the stress gradient, and the residual tensile stress in the unpeened core.

Thus, although the maximum stress in the core when calculated by the stress gradient is apparently less than the surface stress, the maximum core stress, resulting from the combined flexural stresses and residual stresses induced by peening, is greater than the surface stress. When specimens *D*, *E* and *F* are subjected to the bending loads of Fig. 4, the maximum core stress exceeds the surface stress by 1.33; 0.5; and 0.166 percent respectively.

Whether the depth of the peened layer is increased or decreased within the normal range of peening intensity, this relation between maximum core stress and surface stress in bending fatigue specimens is substantially the same.

Shot peening is only one of the cold working processes that induce residual compressive stresses in the surfaces of processed metals. The effects are much the same whether the stress is induced by honing, tumbling, or rolling, except that in honing the induced stress and the depth of the affected layer are considerably less than from ordinary shot peening, and in rolling (Ref. 35) the residual stress magnitude may approach the nominal yield stress of the metal.

Shot Peening More Effective on Hard Steel

Shot peening has been found to be more effective in increasing the fatigue strength of hard surface steel specimens than of softer specimens. By the alternate theory, this phenomenon would require more effective "work hardening" when shot peening steel of Rockwell C 62 than when peening steel of low hardness.

Little if any measurable increase in hardness, however, can be detected after peening very hard steel. Also, as is shown in Fig. 2, the depth of the peened layer in the harder steel is less than in the softer steel. For this reason, the fatigue strength should decrease instead of increase with hardness, since the nominal stress difference between the surface and the unpeened sub-surface metal would be reduced.

These data support the theory that shot peening is effective in increasing the fatigue strength because the residual surface compressive stress offsets, in part, the inherent weakness of surfaces that are subjected to repeated loads.

The theory of increased fatigue strength of surfaces by "work hardening" is further weakened by the relationship between indentor hardness measurements and surface stress shown in Fig. 6. The data recorded in this chart were obtained from steel disks that were deformed from a plane surface toward spherical shapes for the purpose of developing bi-axial stresses in the surfaces.

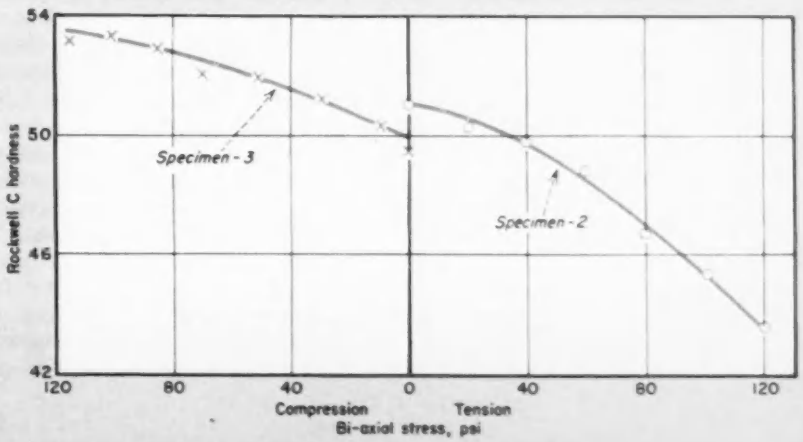
Five Knoop hardness readings were taken at equal angular intervals for each of several stress levels within the elastic range of the specimens. The plotted points are the average of the five readings converted to the Rockwell C scale. Note that compressively stressed surfaces appear to be harder than the unstressed specimens, and that tensile stressed surfaces appear to be softer.

This apparent hardness change is to be expected from instruments that measure hardness by indentations, because such instruments are also crude measuring devices, as was noted by Crampton (Ref. 8) in testing the hardness of drawn brass tubes. Compressively stressed surfaces resist penetration by the indentor more than tensile stressed surfaces do. Since the depths of the indentations are taken as measurements of relative hardness, errors in real hardness in residually stressed metal are unavoidable.

Indentor measurements of "work hardening" must also reflect the magnitude of the residual stress in the metal being measured. It would be expected, therefore, that shot peened and other cold worked specimens would show increased hardness readings in the affected metal and decreased hardness readings in the metal immediately below the peened layer that is residually stressed in tension.

In Fig. 7 are shown the results of a series of hardness readings made on each of twelve shot peened specimens, which were ground and lapped to a 100 to one slope to depths sufficient to penetrate the cold worked metal (Ref. 9). Knoop hardness measurements were made

Fig. 6 — Indentor hardness measurements and surface stress in steel disks that were deformed from a plane surface toward vertical shapes.



at 0.039 in. intervals as measured on the inclined planes, which thus correspond to nominal depth intervals of 0.00039 in. These hardness readings, converted to Rockwell C scale, are averaged in Fig. 7, together with the average hardness readings of the same specimens as taken from non-peened areas.

The peened areas show greater hardness numbers near the surface, where the residual stress is compressive; and lower hardness numbers in the deeper metal, where the residual stress is tensile. The readings taken from non-peened areas show hardness intermediate between those for the peened areas, as would be expected since the residual stress in these areas was presumably zero.

The numerous hardness gradient measurements that are recorded for carburized, nitrided, and other residually stressed surfaces, including hardenability test specimens, give only the indentor readings. These readings are probably in error in regard to real hardness for the compressively stressed surface as well as for the tensile stressed core.

No data are known that attempt to differentiate between "real" hardness and stress hardness, therefore, the extent of real "work hardening" in any specimen is not known.

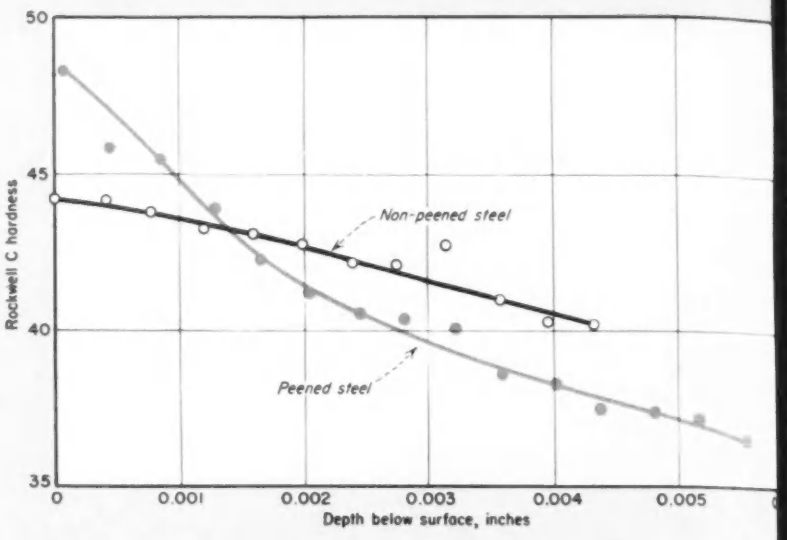


Fig. 7 — Hardness measurements on cross sections of peened and non-peened steel specimens that were ground and lapped to a 100 to one slope to depths sufficient to penetrate the cold worked metal.

Some work hardening change probably occurs similar to that which seems to follow the cold working of copper, as indicated by the apparent change in the elastic limit of tensile specimens when loaded above the proportional limit. With this increase of yield strength under tensile loading, however, there is also a decrease in the elastic limit under compressive loading.

Pull-Pull Fatigue Tests

Another fatigue strength comparison, and perhaps better for indicating surface weakness than specimens tested in bending, is one in which the bending stress gradient as shown in Fig. 4 does not occur, or, at least, is greatly reduced. Data for such a comparison were obtained from a series of fatigue tests in which only tensile loads were repeatedly applied on steel specimens of 0.187 in. dia. The tests were conducted by the Research Laboratories Division, General Motors Corporation. Fig. 8 shows the dimensions of the specimens and the results of the tests.

The specimens were made from new Allison engine connecting rod bolts, AEC 4340-X steel hardened and tempered to 33-36 Rockwell C hardness. Their surfaces were carefully ground and smooth finished in a manner intended to avoid the development of residual stresses of appreciable magnitude. All specimens were subjected to a calculated stress cycle of plus 32,500 psi to plus 150,000 psi. Several specimens were measured to certify that the maximum stress was less than the nominal yield stress of the steel.

The term "nominal yield stress" is used to identify the limit of proportionality, which is the static stress at which the tensile stress-strain curve deviates from a straight line. The term "nominal elastic limit" would serve as well, except that the symbol Y is easily recognizable as indicating yield. The word "nominal" indicates that the elastic limit or the yield stress as determined by static loads do not define the limit of proportionality when the specimen is subjected to often repeated loads.

To repeat a statement made earlier in this article, "... under dynamic loads the elastic limit of ordinary structural materials is so low that plastic deformations, on a macro scale, the extent of which vary with the manner of loading, often alter the stress continuously during a fatigue test."

In the series of tests on the specimens made from the rod bolts, it was desired to conduct the tests at high stress levels to cause early failure of the non-peened specimens. This precaution was necessary to assure that failure of the more durable peened specimens would also occur in reasonable time, since the fatigue strength comparison was to be made on a relative life basis.

At the left-hand side of Fig. 8, the results of tests on fifteen non-peened specimens are shown; each bar represents one specimen and its height represents the number of cycles that were applied before failure. These bars are arranged in ascending order for easy comparison.

In Fig. 8 are also shown the results obtained from tests of three groups of shot peened specimens, which differed from one another only in the intensity of the peening

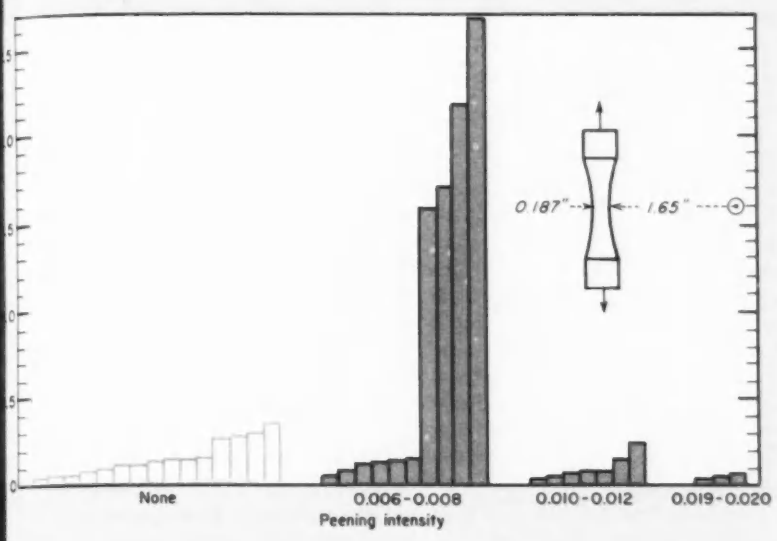


Fig. 8 — Relation between tension fatigue durability and shot peening intensity, as indicated by pull-pull fatigue tests on AEC 4340-X steel specimens hardened and tempered to 33-36 Rockwell C hardness.

that was applied. In arranging the tests it was assumed that light peening would prolong the life of the specimens, because the surface weakness would be overcome by a very thin layer of residual compressive stress without seriously increasing the sub-surface tensile stress. It was also believed that more intense peening could reduce the fatigue durability because of the greater internal tensile stress as well as the more severe bruising of the peened surface.

Light Peening Superior to Heavy Peening

From Fig. 8 it is seen that the durability of the lightly peened second group of ten specimens was greater than the non-peened specimens, and that the durability decreased as the peening intensity was increased. The data suggest that peening lighter than the 0.006 to 0.008 A2, which was used on the second group of specimens, would have given even better fatigue strength and perhaps reduced the variability of life among the specimens.

Since the diameters of these specimens were 0.187 in., the residual tensile stress in the unpeened core metal was relatively great, as is indicated in the diagram *D*, Fig. 5. The internal residual tensile stress was added to the nominal tensile stress from the external load. Varying the peening intensity does not greatly alter the magnitude of the residual compressive stress in the surfaces; the effect being mainly to alter the depth of the peened layer. It is to be expected, therefore, that peening pull-pull specimens to a greater depth than is necessary to counteract surface weakness reduces their fatigue durability by increasing the internal residual tensile stress, and thereby increases the maximum test stress.

It is probable that for the pull-pull specimen shown in Fig. 8, the depth of the peened layer could have been reduced to not more than 0.002 in. with actual increase in fatigue strength, as compared to the greater but unmeasured core residual tensile stress that resulted from the peening intensity that was applied. The loss of fatigue strength for the third and fourth groups is presumably chargeable to greater core residual tensile stress, together with the more severe bruising of the surfaces of the specimens that accompanied the greater peening intensity.

The results of these tests, particularly the qualitative fatigue strength comparisons of the non-peened and the lightly peened groups of specimens, favor the theory of surface weakness. Quantitative fatigue strength comparison cannot be obtained from such tests, even if the actual stresses at the surface and in the sub-surface metal were known, because so far as could be determined by visual inspection of the fractures, no failures originated in the sub-surface material.

To measure the relative strengths of surface and sub-surface metal it is, of course, necessary to know the stress causing fatigue fractures in each of the regions where fatigue strength is to be compared. It is also important to conduct the test in such manner that the fractures for each region to be compared develops in approximately the same number of stress applications, or that a sufficient number of specimens are tested to establish the fatigue endurance limit.

In the process of shot peening, the peened surfaces are severely bruised and notched by the shot impacts. When chilled iron shot are used many of the shot are fractured. Visual inspection will reveal numerous sharp cuts and an occasional instance in which fractured shot fragments are embedded in the peened surface.

Surface Imperfections Are Stress Raisers

All these added surface imperfections constitute stress raisers of the same severity as if they occurred in unpeened surfaces. They will augment the damaging tensile stress from the external load in the same manner that any form of surface imperfection raises the magnitude of the applied tensile stress. As long as the residual compressive stress, however, is greater than the local increase in tensile stress, as represented by the most severe stress raiser, the fatigue strength of the specimen will be increased by the peening.

Residual Stress Measurements

To discuss the effects of residual stresses on fatigue strength is useless, of course, unless it is first contrived to obtain reasonably accurate measurements of the magnitudes and arrangements of the principal stresses.

The residual stresses in planes of the principal stresses of a large variety of specimens have been measured by the General Motors Research Laboratories Division using the dissection method.

This process, which has been used since it was first reported by Heyn (Ref. 10), consists of removing successive layers of material from the specimen and measuring the deformation of the remainder of the specimen after each layer is removed. From these measurements, the original residual stress in the specimen is calculated.

The principal interest is in measuring, as accurately as possible, the residual stress in the plane of the principal stress as identified by the direction of the applied load and the direction of the fatigue fracture. Since the residual stress gradient is often very steep, it is important that the thickness of the successive layers removed be as thin as possible.

In the process of removing metal, it is easily possible to introduce new residual stresses as great or greater than the original stresses being measured. This danger increases as the thickness of the removed layer is decreased. Since the maximum thickness of successive layers near the surfaces and elsewhere, when the rate of stress change is large, must be of the order of 0.0005 in., each step in the process requires the utmost care and precision. For example, a layer of 0.0005 in. is removed in steps not greater than 0.0001 in. For a thicker layer, the final 0.001 in. of metal is also removed in not less than ten approximately equal steps.

Since the forces and force moments of the original residual stresses in the specimen were in equilibrium, a check upon the accuracy of the dissection stress measurements can be made by calculating the degree of unbalance of the forces and force moments from the reconstructed stresses. Errors are found to range up to plus or minus five percent.

It should not be understood that the errors in balance of forces and force moments indicate that the maximum errors in the reconstructed stresses are within five percent of their actual values. It is probable that, in regions of rapid stress changes, particularly near the surface, the errors in stress magnitude may be considerably greater because only a small portion of the original specimen is used for the measurements.

Since the stress in any specimen is probably never more simple than biaxial, losses of residual stress will inevitably occur when a small dissection specimen is removed from a large sample. Errors also arise from the fact that the effect of stresses in planes other than the one being measured are neglected. Although a high degree of accuracy in stress magnitude is desirable, it is often more important to know accurately the pattern or arrangement of the residual stresses within the specimens.

By dissection processes, the residual compressive stress that is induced by shot peening has been found to approximate one-half of the nominal yield stress of the peened metal (Ref. 11), as is indicated in the chart Fig. 2. Dissection stress measurements, however, are not capable of detecting highly localized stress variations. It is probable that the stress induced by each shot impact is not uniform in all parts of any single indentation, therefore, it is to be expected that the minimum residual stress in a shot peened surface will be less than the measured average. Added to this uncertainty is the probability of micro stresses originally in the metal, and the unknown magnitude of the individual stress raisers that are formed by the peening.

Greater gains in fatigue strength result when the peening is performed with whole shot, which are substantially spherical, than when the peening material is contaminated with broken and deformed shot (Ref. 12). This gain presumably results from the avoidance of cuts and abrasions, and the consequent reduced severity of the peening stress raisers with, perhaps, more uniform intensity and coverage because of the uniformity of particle size.

Stress From Shot Peening Controllable

For the foregoing and other reasons, the increased fatigue strength that is obtained by ordinary shot peening can be used only as a qualitative measure of surface weakness. A somewhat better appreciation of the magnitude of surface weakness can be had by increasing the mechanically induced surface residual compressive stress sufficiently to cause fatigue fractures to originate in sub-surface metal. Such an increase in surface stress can be accomplished by "Strain Peening," a process that was first used by the author (Ref. 13) to prevent stress corrosion cracking.

When shot peening has been applied to attain adequate coverage, regardless of the state of residual stress that prevailed in the surface before peening, the average induced stress is not greatly increased with increased time of exposure of the surface to the impacting shot, or with the size, or the velocity of the shot. The effect of these variables is mainly to vary the depth of the residually stressed layer.

If before peening the surface is residually stressed in tension, as it will be if finished by grinding, the magnitude of the compressive stress induced by peening will be the same as that which would be induced if the surface had been initially stressed in compression, or if it had been entirely free from stress.

By applying shot peening, or an equivalent operation, however, while the surface is in a suitable state of strain, the magnitude of the final residual stress after peening can be controlled to equal any value within the range from the nominal compressive elastic limit to approximately one-half of the nominal tensile elastic limit.

Strain Peening

If the tubes shown in Fig. 9 are shot peened while the bolt and tap bolts are slack the tube surfaces will, as in all peening operations, be residually stressed in compression to approximately one-half of their nominal yield stress. Since the specimens are cylindrical, the residual stress will be longitudinal, tangential, and radial; not biaxial as in plane surfaces. The longitudinal and tangential stresses will approximate one-half of the nominal yield stress of the metal.

Suppose the bolt, Fig. 9 (*A*), is tightened to stress compressively the tube to one-half of its nominal yield stress. (The stress from the load applied by the bolt acting in the tangential and radial directions will here be neglected.) If the tube is shot peened while loaded, its surface while loaded after peening will be residually stressed to the same extent as if no external load was present, that is, the longitudinal and tangential stress will be one half of the nominal yield.

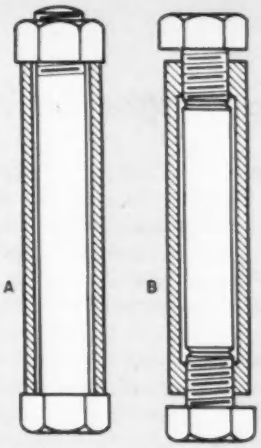


Fig. 9 — Tubular specimens adapted for strain peening. — (A) Tube pre-loaded in compression.— (B) Tube pre-loaded in tension.

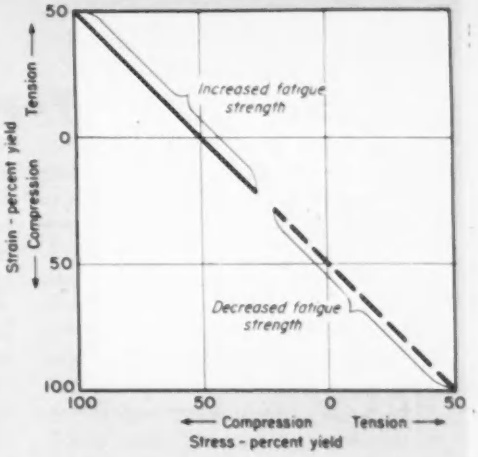


Fig. 10—Approximate residual stress resulting from strain peening in percent of nominal yield stress. Possible residual stress ranges from a yield stress of 50 percent in tension to 100 percent in compression.

RESIDUAL PEENING STRESS LOST. Removal of the axial load by releasing the bolt will permit the tube to lengthen elastically to nearly its original dimension. As the tube lengthens, the residual peening stress acting in the axial direction will be dissipated and only the tangential residual stress will be retained.

If the service or test loads applied to the tubular specimen are to act principally in the longitudinal direction, the retained tangential stress will have little effect on the fatigue strength of the specimen. The peening, however, will seriously diminish the fatigue strength of the tube against repeated loads acting in the axial direction. The surface will be cold worked to approximately the same degree as in normal peening, and stress raisers of similar severity will be formed, but these effects serve only to increase the weakness of the surface. The immunizing effect of shot peening is lost with the loss of the residual compressive stress.

The extent of damage from shot peening will increase as magnitude of the compressive bolt load, at time of peening, is increased.

For example, if the applied bolt load is increased to stress the tube to its nominal compressive yield strength and the specimen is then shot peened, the residual stress at the conclusion of peening will as before be equal to one-half the nominal compressive yield stress. During peening the surface compressive stress is decreased from the 100 percent stress imposed by the bolt to the 50 percent yield stress of normal peening.

Upon release of the bolt the tube will elastically recover nearly its original length, whereby the surface will be extended to alter the stress from one-half nominal compressive yield to nearly one-half nominal tensile yield. The loss in fatigue strength will now be greater because not only is the surface vulnerability increased by stress raisers, but the damaging tensile stress from the service load is increased by the surface residual tensile stress.

PEENING STRESS DOUBLED. Beneficial strain peening results when suitable specimens are shot peened while stressed in tension. In Fig. 9 (*B*) is shown a tubular specimen with a tap bolt fitted in each end. When tightened the tap bolts bear upon a strut within the tube. By tightening the tap bolts against the strut, the tube can be loaded in tension to any desired stress.

If the tube is stressed to one-half its nominal tensile yield stress by tightening the screws and then shot peened, the peening will change the surface stress from tension to compression. The residual stress magnitude, as in ordinary peening, will be one-half nominal compressive yield stress.

Upon release of the tap bolts the tube will contract to nearly its original length. The contraction will not greatly alter the tangential residual stress, but will increase the longitudinal residual compressive stress from one-half nominal yield stress to double that amount. As a result of the increased residual compressive stress, greater protection will be available against surface weakness, and the fatigue strength of the specimen under repeated longitudinal tensile stresses will be greatly enhanced in any test that does not dissipate the residual surface stress.

PEENING STRESS VARIABLE OVER WIDE RANGE. The range of residual stress, in terms of nominal yield stress, that can be induced in the surface of shot peened specimens by strain peening is shown in Fig. 10; the possible residual peening stress ranges from 50 percent yield stress in tension to 100 percent yield stress in compression.

For increasing fatigue strength, practical peening is limited to the solid line, Fig. 10. The broken line is of value in providing specimens, whereby the relative effectiveness of "work-hardening" and residual stress may be measured.

Idealized Strain Peening Diagrams

Diagram Fig. 11 (*A*) shows a beam loaded in bending by a downward acting load at *O*. The resulting stress magnitude at any depth within the beam is represented by the horizontal distance from the vertical line OO' to the diagonal line N_cN_t. The externally applied stress ON_c on the upper surface is one-half the nominal compressive elastic limit* OY_c; and the tensile stress $O'N_t$ on the lower surface is one-half the nominal tensile elastic limit $O'Y'_t$.

When both of the strained surfaces of the loaded beam, Fig. 11 (*A*), are shot peened, the usual compressive residual stress is induced in the transverse direction; but the peening does not alter the longitudinal stress on the upper surface, since this stress is already compressive and is approximately equal to the stress that would otherwise be induced by peening.

Since shot peening always induces bi-axial stresses, and transverse stress is relatively unimportant in the bending specimens being discussed, hereafter reference to stress will relate to longitudinal stress.

The stress in a thin layer on the lower surface, Fig. 11 (*A*), which is stressed in tension by the applied load, becomes compressive after peening by the amount $O'M$, which is equal to one-half the compressive elastic limit $O'Y'_c$.

When the external bending load is removed, elastic recovery of the beam removes all except the stress change that was induced in the beam by the peening, as shown in Fig. 11 (*B*). This stress change is equal to the distance N_tM, Fig. 11 (*A*), measured from the vertical line OO' of Fig. 11 (*B*). That is, the residual compressive stress on the lower surface of the beam is now equal to the distance $O'M'$, which is also the nominal elastic limit of the metal. There will be a permanent set in the free beam because of the residual compressive stress on the lower surface.

In resisting repeated loads applied in a downward direction, the fatigue strength of this strain peened beam will be greater than is obtainable from simple peening. The greater residual compressive stress on the lower side of the beam will reduce the magnitude of the surface tensile stress from the downward acting loads. However, the fatigue strength of this strain peened beam against repeated loads acting in an upward direction will be decreased, since there is no residual compressive stress to offset the stress raisers that were formed on the upper surface by the peening.

If at the time of peening, the specimen is strained in compression substantially greater than one-half the nominal yield strength, for example such that the applied stress equals the elastic limit as shown in Fig. 12 (*A*), the compressive stress on the upper surface will be reduced by the peening to the distance *OJ*, which is approximately one-half the nominal yield stress.

The resultant residual stress on the upper surface immediately upon removing the external load will be tensile, Fig. 12 (*B*), by the distance OJ', which is equal to approximately one-half the tensile yield stress. The tensile stress on this surface from upward acting loads will then be the sum of the residual stress and the stress from the externally applied load. Since all fatigue fractures are caused by tensile stresses, the result will be extensive loss in fatigue strength because of the combined tensile stresses from the external load, and from the residual tensile stress that was induced by the peening.

PRESTRESSING MAY BE OVERDONE. On the lower surface of the diagram, Fig. 12 (*A*), the sum of the tensile stress from the external load and the residual compressive stress from peening is indicated by the distance Y'_tM. Upon removal of the external load, the tensile stress from the load is removed from the sum and the residual compressive stress equals the distance $O'M''$, as shown on the lower surface of the diagram Fig. 12 (*B*). However, this sum exceeds the nominal yield strength of the metal by the distance $M'M''$, and this portion is therefore immediately dissipated by plastic yielding. The final residual compressive stress will therefore be equal to the yield strength of the metal, exactly as occurred in Fig. 11, which was peened at one-half of the strain that was applied in the Fig. 12 specimen.

The yielding represented by $M'M''$ can be detrimental to the fatigue strength of the specimen, since this stress is first induced by peening, and then lost by compressive yielding. It is possible damage will result by exhausting the capacity of the metal for plastic adjustment through excessive cold working.

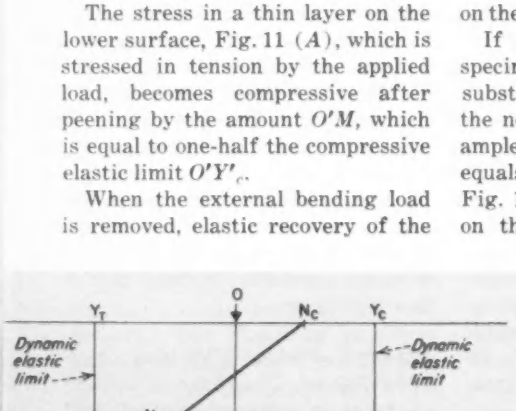
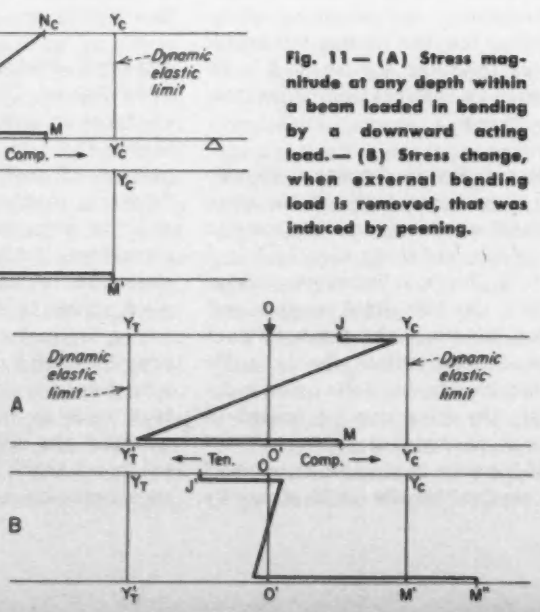


Fig. 11 — (A) Stress magnitude at any depth within a beam loaded in bending by a downward acting load. — (B) Stress change, when external bending load is removed, that was induced by peening.



g. 12 — (A) Stresses in oned beam with applied ess equal to the elastic it. — (B) Resultant resid- l stress when external ad is removed.

*The magnitude of plastic yielding under repeated stresses, particularly when the load is applied only in one direction, is greater than the plastic yielding measured by static loads. Measurements indicate (Ref. 1) that plastic yielding continues at a decreasing rate throughout uni-directional fatigue tests. The stress indicated by the dynamic elastic limit is therefore lower than the conventional yield stress but it does not identify a limit of plastic yielding.

Measurements of Residual Stress

The ability to control the magnitude of the residual compressive stress, by performing the peening while the specimen is strained in tension, supplies a method for estimating, with reasonable accuracy, the relative fatigue strength of surface and sub-surface material. By varying the magnitude of the residual stress, fatigue fractures can be made to originate at the surface or in subsurface metal with substantially equal frequency and after approximately the same number of load applications. Since it is necessary to avoid any loss of protective stress, the surface that is residually stressed in compression must be loaded only in tension.

In experimental work, this condition is most conveniently accomplished by the use of specimens subjected to repeated bending loads acting only in one direction. Dissection stress measurements may then be made to establish the magnitudes of surface and sub-surface residual stresses. These measured residual stresses are added to nominal calculated stress from the external load to give a reasonable measure of the actual stress distribution at the time of fracture.

Such residual stress measurements have been made on specimens, similar to Figs. 11 and 12, except that the specimens had been shot peened on the tension surface only while strained in bending. Fig. 13 shows the residual stress, as measured by the dissection method, developed by strain peening the surface of a leaf spring specimen made from SAE 5150 steel heat-treated to a hardness of Rockwell C 47. The peening was applied on the lower surface while this surface was loaded in bending to a specified tensile strain of 0.00586 in. per in. In terms of stress, this strain is equal to approximately 170,000 psi.

The nominal yield stress of this steel was not measured, but to conform to the expected characteristics of steel of 47 R C hardness it will be assumed to be 200,000 psi, from which the average residual compressive stress induced by shot peening alone may be assumed to be 100,000 pounds per square inch.

From the reconstructed stress diagram, Fig. 13, it is seen that the residual compressive stress on the lower surface is 215,000 psi. This value is the sum of the stress induced by shot peening and the stress caused by the elastic recovery of the specimen after removal of the bending load, which was applied during strain peening. Since the elastic recovery was not complete, the residual stress added by the elastic recovery was less than the 170,000 psi that represented the strain peening load. Because of the plastic changes that occurred during the peening and during the subsequent fatigue test, a permanent set remained in the specimen similar to that indicated by the inclined lines in the unloaded specimens shown in the Figs. 11 (*B*) and 12 (*B*) diagrams.

Permanent Set in Prestressed Specimens

The magnitude of the permanent set remaining in the spring specimen of Fig. 13 is shown in Fig. 14. In this diagram the portion of the line representing residual stress, which lies between the points *C* and *D*, represents metal that experienced no plastic yielding during peening.

Before peening, the line *CD* was coincident with a portion of the vertical line *OO'*. The extent of its failure to return to its original position after peening, and after release of the external load, provides a measure of the permanent set of the specimen. Extending the line *CD* to the upper surface at *A*, and to the lower surface at *B*, permits measurement of the permanent set at both surfaces. On the lower surface this permanent set is represented by *BO'*, a distance equivalent to 85,000 psi tensile stress.

The residual compressive stress that was added to the shot peening stress was therefore 170,000 minus 85,000 or 85,000 psi. The assumed 100,000 psi residual stress from peening plus 85,000 from elastic recovery, equals a total surface residual compressive stress of 185,000 psi. But this value is 30,000 psi less than the residual stress that was measured by the dissection process, which it will be recalled was found to be 215,000 psi as shown in Fig. 13. The greater part of this discrepancy is probably in the assumed magnitude of the elastic limit of the peened metal.

The elastic limit of the surface metal was presumably increased by the plastic yielding that is seen to occur during strain peening in any stress-strain diagram. Through a considerable range of strain, yield strength increases with each of a succession of static tensile tests in which the applied stress exceeds the limit of proportionality.

Strain Peening Stress Increased by Subsequent Cold Work

Dissection measurements were made after the specimens had been subjected to 140,000 cycles of bending

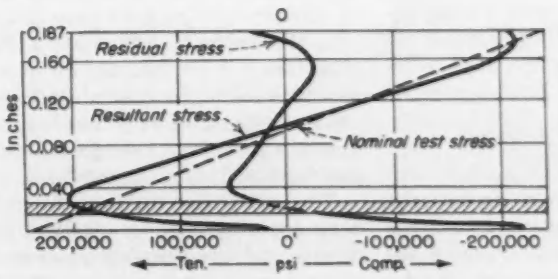


Fig. 13 — Measured residual stresses in a strain peened steel specimen after 140,000 stress cycles; specimen was shot peened only on the tension surface while strained in bending.

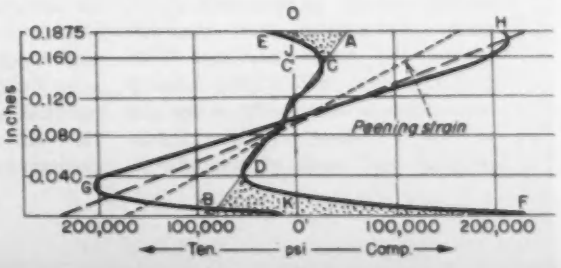


Fig. 14—Increase in fatigue strength by plastic yielding in specimens under unidirectional loads.

stress by downward acting loads. The dotted line Fig. 14 indicates the strain that was applied to the specimen during the peening. Since the fatigue test loads applied nominal stresses ranging from plus 25,000 to plus 240,000 psi, additional local plastic adjustment of the metal occurred in the most highly stressed regions during the test.

The elastic limit indicated by the dissection measurements is presumably a close approach to the dynamic elastic limit of the metal. Since the static elastic limit exceeds the dynamic elastic limit, and since the maximum measured residual compressive stress near both surfaces is 215,000 psi, it appears that the assumed nominal yield stress and the assumed residual stress induced by peening should be revised upward.

In Fig. 14 it is seen that extensive plastic yielding occurred to considerable depths from both surfaces. The extension of the line *CD* to the upper surface at *A* encloses the triangular darkened area *AECA*, which represents plastically deformed metal. The extent of yielding at any depth is, of course, the horizontal distance over the darkened area. The maximum yielding occurred at the surface, and approximates 83,000 psi. The horizontal distance from the vertical line between the points *C'* and *O* to the resultant stress line *HGK* is a measure of the compressive dynamic elastic limit of the steel near the upper surface of the specimen. This limit is seen to vary from 215,000 to 180,000 psi, the latter value presumably being a close approach to the dynamic elastic limit of that part of the specimen metal that had not been "cold worked."

Residual Stress Increased by Dynamic Yielding

Similar measurements can be made of the plastic movements of the metal near the tensile stressed under side of the specimen in the darkened area *BDFB*. The yielding in this area is complicated by the plastic movements that occurred during the strain peening. It is probable that all of the plastic movement in the metal near the surface is a direct result of strain peening but most of the plastic movement of the metal at depths greater than 0.020 in. was probably caused by dynamic yielding.

It may be, however, that some deep yielding occurred during the strain peening. Although the surface stress was presumably less than the static elastic limit, it is nevertheless possible that yielding occurred because of the severe vibrations from the impacts of the shot. Since plastic yielding also occurred at depths of 0.025 in., where the stress during peening approximated 125,000 psi, most of the deep yielding probably occurred during the early part of the fatigue test with slow plastic movements (creep) continuing for the duration of the test. The resultant test stress at the limiting depth of plastic yielding is seen to be 190,000 psi approximately.

Prestress May be Lost

As a result of the yielding shown in Fig. 14, the fatigue strength of this specimen is greatly increased in any repeated load test that does not dissipate a substantial portion of the residual stress. When the residual compressive stress is made equal to the dynamic elastic limit of the metal, any additional compressive stress from external sources will necessarily cause compressive yielding, and thereby reduce the protection that is bestowed to the weak surface by the residual compressive stress.

For example, any upward acting bending load on the specimen Fig. 12 (*B*) will cause compressive stress on the lower surface. Since the compressive stress on this surface is already equal to the elastic limit, any additional stress in the same direction will necessarily result in plastic yielding, and thus dissipate residual stress in an amount equal to the stress from the upward acting load.

The maximum gain in fatigue strength that may be obtained by mechanical prestressing processes, therefore, is limited to specimens that are compressively prestressed to the elastic limit of the metal, as shown in Figs. 11 (*B*), 12 (*B*), and 13, providing they are repeatedly loaded in bending in a downward direction only. Any reversal of load will reduce the residual compressive stress that protects the weak surface against fatigue fractures.

The specimen shown in Fig. 13 was repeatedly loaded by a downward acting load applied as indicated by the arrow. This load exerted a maximum nominal surface stress at each load application of 240,000 psi, as shown by the broken diagonal line. The actual stress, shown as "Resultant Stress," is the sum of the nominal stress and the dissection measured residual stress. Under the nominal applied stress of 240,000 psi, the maximum compressive stress of 215,000 psi occurs about 0.010 in. below the upper surface, and the maximum tensile stress of 205,000 psi occurs at 0.030 in. from the lower surface.

Dynamic Yielding Increases Fatigue Strength

The dynamic yielding, which occurs during fatigue testing under unidirectional bending loads and unidirectional twisting loads, develops residual stresses that reduce the actual stresses in the specimens. As seen in Fig. 14, the sub-surface yielding reduced the stress magnitude. It is well-known that the fatigue strength of conventional non-prestressed specimens is greater when they are repeatedly loaded in one direction only than when they are subjected to reversed loads.

The measured residual stress on the non-peened upper surface of the beam, Fig. 13, which is nominally stressed only in compression, is actually a tensile stress. The measured tensile stress on this surface is greater than the maximum tensile stress on the peened lower surface, which is nominally stressed only in tension. The residual stress on the non-peened upper surface as has been described, is the result of compressive plastic yielding of this surface.

Fatigue failures frequently develop on the "compressively" stressed side of fatigue specimens. In specimens that are presumed to be free from residual stresses, such "compressive" failures are always caused by tensile stresses that are developed from compressive yielding. In the absence of adequate residual stress measurements, however, they are erroneously believed to be cohesive failures that result from compacting stresses.

Fig. 15 — Sub-surface fatigue nucleus close to the tension surface of a prestressed spring specimen.

Prestressing for Efficient Use of Material

In specimens that are correctly prestressed for the most efficient use of material to support repeated bending loads in one direction only, such as is approached by the specimens Figs. 13 and 14, fatigue failures may originate in any of three locations. In the discussion of these three possible sources of fracture, it will be assumed that the residual stress measurements are exact.

1. The fracture could originate on the upper surface from the tensile stress OE. Fractures in this region would not develop to complete failure because the crack would propagate only to the depth OJ, which is the limiting depth of the tensile stress. The sub-surface compressive stress would form an impassable barrier to further progress of cracks. However, in the specimen illustrated, fractures could not originate on the upper surface because the stress OE can exist only when the external bending load is completely removed. Since the nominal downward acting bending stress during the fatigue test ranged from 25,000 to 240,000 psi, and since the measured tensile stress OE is 28,000 psi, the maximum tensile stress on the upper surface could not exceed 28,000 minus 25,000 or 3,000 pounds per square inch.

2. Fatigue fractures could and did originate on the lower surface from the tensile stress $O'K$, which is equal to 20,000 psi. The effective stress was greater than the dissection measured value because of the many severe stress raisers that were formed by the shot during peening.

3. Fatigue fractures could and did originate in sub-surface metal at the depth G from the tensile stress $O'G$, which is equal to 200,000 pounds per square inch.

In Fig. 15 is shown a fatigue fracture of sub-surface origin that developed in another of the group, not the one represented in Figs. 13 and 14, of prestressed spring specimens that were fatigue tested.

Sub-Surface Fractures in Efficient Specimens

Sub-surface fractures from tensile stresses of 200,000 psi occurred with approximately the same frequency as the surface fractures from tensile stresses of 20,000 psi. The fatigue strength of the surface was therefore one tenth of the fatigue strength of sub-surface metal. This strength ratio exaggerates the weakness of the surface because of the presence of stress raisers. Also the dissection stress measurements, particularly the surface stresses, are not sufficiently reliable to warrant the use of precise numerical comparisons.

Tests are contemplated on other specimens identical with the specimen shown in Fig. 13, except that the surface will be polished. The polishing will be applied to strain peened specimens and to specimens that have not been prestressed. After polishing, one group of non-prestressed specimens will be tempered in a vacuum or in a neutral atmosphere to reduce the residual compressive stresses that are induced by the polishing-operations. The results of such tests should provide a better measure of surface weakness than the data given in Fig. 13, because removal of the obvious stress raisers will remove most of the extra surface hazards introduced by the peening.

Dissection Measurements Require Great Accuracy

The cross-sectioned band near the lower surface of the beam, Fig. 13, indicates the portion of the specimen that remained when the dissection measurements were completed. Its thickness is 0.0105 in. During the dissection twenty layers were removed from the upper side of the beam and thirteen layers were removed from the under side. The dissection proceeded from the upper side to within 0.026 in. of the opposite surface before any metal was removed from the lower surface. In this manner, the accuracy of the measurements of stress near the critical tensile stressed surface was increased. Calculations of the moments from the measured residual stress show that the error in the reconstructed stress diagram is 3.4 percent.

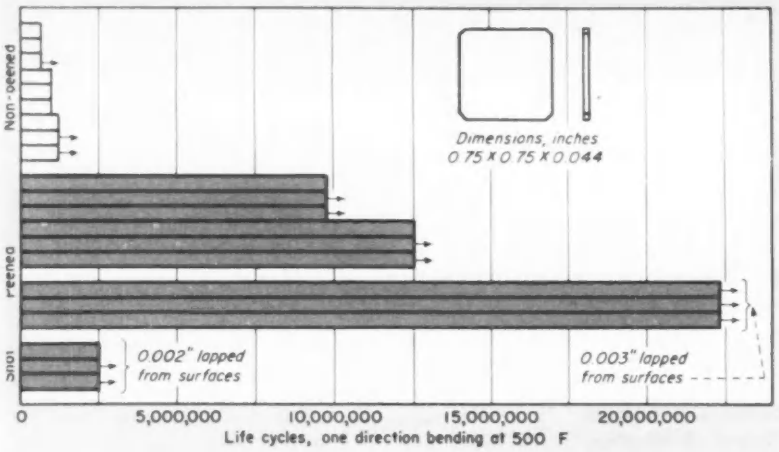
Effect of Removal of Stress Raisers

A few tests have been made in which shot peened specimens were polished after peening. The magnitude of the residual stress was presumably not altered, but the polishing served to remove most of the obvious stress raisers that were left by the peening.

A series of tests conducted by Wright Field, and reported by the Office of Scientific Research and Development (Ref. 7), compared the fatigue strength of shot peened specimens with specimens that had been polished after peening. The latter showed an increase in fatigue strength of 20 percent at the endurance limit. It is improbable that the polishing added beneficial residual stresses, since the surface prestressing from peening was still effective. The gain probably results from the greater effectiveness of the peening stress because of the removal of obvious stress raisers.

Similar results were obtained from an earlier series of tests by Wright Field, which also are reported (Ref. 14) by the Office of Scientific Research and Development.

The results of an interesting se-



NDRC Report OSRD-4825, Serial No. M-476.

Fig. 16—Fatigue durability of non-peened and shot peened air compressor discharge valves that were surface finished by several methods.

ries of tests, also reported by the Office of Scientific Research and Development (Ref. 15), that included honing after peening are shown in the bar chart, Fig. 16, which is replotted from the original report. The chart compares the fatigue durability of non-peened and shot peened air compressor discharge valves, made of SAE 1080 steel heat-treated to Rockwell C hardness of 46-52, that were surface finished by several methods. Dimensions of the valves are shown in the chart. Shot peening was done by the General Motors Research Laboratories Division to an intensity of 0.007A2. The test valves were installed in air compressors and life tested, by Bendix Westinghouse Automotive Air Brake Company, with the results shown in the chart.

Excessive Honing Destroys Peening Effect

The increased durabiilty resulting from shot peening, as seen in Fig. 16, was further enhanced by lapping 0.002 in. from the peened surfaces. This amount of lapping was sufficient to remove the peening stress raisers. Because of the low intensity peening applied to these hard specimens, and the consequent rapid diminution of residual stress with depth, it is probable that the magnitude of the residual stress was also reduced by the relatively deep honing.

The beneficial effect of removing the peening stress raisers more than offset any loss of residual stress, since no failures of the peened and honed specimens occurred in the test, which was stopped after more than 20,000,000 stress cycles. Note, however, that when 0.003 in. was lapped from the valves, the compressively stressed layer from peening was penetrated; in consequence of which the fatigue weakness of the surface reappeared and the fatigue durability dropped to only slightly better than honed, but non-peened valves.

Residual compressive stress can be induced by other mechanical processes such as tumbling, hammering (Ref. 16), burnishing, and superficial rolling (Ref. 17). Rolling presents several advantages because: (a) The depth of the induced stress is controllable and reproducible, (b) excessive local cold working can be avoided, (c) the operation does not introduce severe stress raisers, and (d) the induced residual compressive stress is more uniform and of greater magnitude than can be induced by individual impacts such as shot peening. The presumed superiority of rolling, however, has not been proved since adequate comparative fatigue strength measurements have not been made.

Rolling is not so versatile as shot peening or tumbling, since it cannot be applied to irregular shapes, but by the use of suitable tools rolling may be applied to symmetrical shapes such as plane surfaces, cylindrical shafts and bores, and other surfaces of revolution.

In many of these, strain rolling can be used to increase the residual compressive stress in the same manner as that described for shot peening. For example, a shaft or tube such as the strained specimen shown in Fig. 9 can be burnished, tumbled, hammered, or superficially rolled while stressed in tension. The tensile strain can be obtained by axial loading; or the rolling or other prestressing tool can be applied on the tension side of a shaft that is strained by a bending load, such as is applied to a rotating beam fatigue specimen.

Strain peening, strain rolling, or strain prestressing by other operations including polishing, can be advantageously applied to torsion bars that are to be repeatedly twisted in one direction only. The bar being prestressed is strained in the direction of the normal load while the rolling or peening is applied. Something less than the maximum prestress can be retained when low magnitude twisting loads on torsion bars, or bending loads on leaf springs, occur in the opposite direction to the normal working load. Reversed loads at relatively low stress often occur in vehicle suspension springs from rebound.

Polishing Fatigue Specimen Defeats Purpose

As a part of his investigation of the effect of specimen size on fatigue strength, H. F. Moore tested three groups of rotating beam specimens made from SAE 1035 steel finished by different processes (Ref. 18 and 19). The first group of specimens was tested in the as rolled and polished state. The second group was the same as the first except that the specimens were bright annealed in vacuum after polishing. The third group was the same as the second except for repolishing after being vacuum annealed. These tests did not supply important data for size effect studies. But when interpreted in terms of surface weakness and the effect of residual stress on fatigue strength, these data become very important.

Specimens of three different diameters were tested in the first and second groups, but only one diameter of specimen was tested in the third group. Annealing was accomplished by holding the specimens at 1500 F for two hours followed by slow controlled rate cooling. The annealing did not alter the

surface finish, but it presumably removed any residual compressive stress that was induced by the polishing operation. The comparison shown in Table I is the fatigue strength in pounds per square inch at the endurance limit and the percent gain in fatigue strength that was caused by the residual compressive stress induced by polishing.

The specimens in group 2, which were annealed after polishing, are definitely weaker than the specimens shown in group 1 that were not annealed. The specimen in group 3, which was repolished after annealing, has recovered the same endurance limit strength as the original "as rolled and polished" specimens although the surface smoothness was not altered.

Residual Stress Induced by Various Operations

The greater strength of the specimens in group 1 and 3, Table I, results from the surface residual stress that was induced in a very shallow layer of the specimens by the polishing operation. That the residual stress from polishing is compressive is shown by the three plates of Fig. 17, the upper surfaces of which were finished by the processes indicated.

Each of the three plate specimens was straight and relatively free from residual stress before the upper surface finishing operation was applied. The curvatures shown, therefore, result from the stress developed in the upper surface of each strip during the finishing. Note that specimen *A* is curved concave on the upper surface in response to the tensile stress developed by grinding. Specimens *B* and *C* are curved convex on their upper surfaces, showing that honing and filing develop residual compressive stresses. Dissection stress measurements (Ref. 11) indicate that the initial residual compressive stress from polishing may exceed 20,000 pounds per square inch.

A file finish must not be assumed to be as effective as finishing by honing, even if the residual compressive stresses should be of the same depth and magnitude. The surface scratches produced by filing constitute stress raisers, and the effect of the compressive residual stress is reduced by the augmented tensile stress from these stress raisers.

It is probable that grinding induces residual compressive stress by the plastic "smearing" of the metal, similar to that which occurs in honing and filing. However, since the rate at which metal is removed generates heat, and since highly localized heat develops residual tensile stress, the net effect of ordinary grinding is tensile stress.

Experiments by General Motors Research Laboratories have shown that thin stress free plates ground on one side may indicate compressive stress or tensile stress depending upon the grinding wheel characteristics and the rate at which metal is removed. Compressive stress is indicated when the specimen is ground on one side using a soft, sharp, coarse grit wheel and removing shallow cuts of 0.0001 in. or less per pass. In practical grinding operations the residual stress is always tensile, and ground surfaces are therefore always weaker in fatigue than compressively stressed surfaces of equally smooth finish. High speed polishing by abrasive covered felt wheels is similar to grinding and therefore often produces inferior parts.

Fatigue Test Data May Be Misleading

From the H. F. Moore tests, it may be concluded that the practice of low speed polishing of fatigue specimens not only removes obvious stress raisers, but it also imparts additional strength by virtue of inducing residual compressive stress.

This increased strength destroys the accuracy of the test in so far as the test reflects the fatigue strength of the specimen material. For this reason few if any fatigue tests of "ideal" highly polished specimens, one favorite form of which is shown in Fig. 18; have fulfilled their purpose. The standard fatigue specimen, Fig. 18, is included here to emphasize that standard procedure merely perpetuates the errors that are inherent in

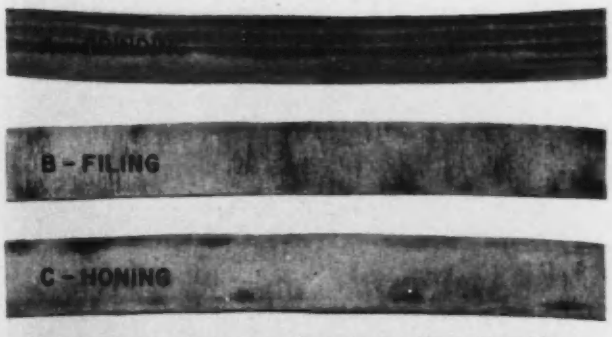


Fig. 17 Above — Curvature of specimens shows stress induced by various treatments. (A) Tension. (B) and (C) Compression. Fig. 18 Below — Standard rotating beam fatigue specimen gives misleading results.

Table I — Fatigue Strength at the Endurance Limit Of SAE 1035 Steel Rotating Beam Specimens

Group	Condition of Specimen	Specimen Dia, In.		
		0.125	0.250	0.500
1	As rolled and polished	39,000	39,000	35,000
2	As rolled-polished-vacuum annealed	35,000	34,000	31,500
3	As rolled-polished-vacuum annealed-repolished	.....	39,000	.....
	Percent fatigue strength increase resulting from residual compressive stress	11	15	11

the specimen preparation and in the manner of loading. The magnitude of the errors no doubt vary considerably depending upon the manner in which the specimens are tested, and the initial magnitude of the compressive stress that was induced. For the H. F. Moore specimens the polishing residual stress, given in Table I, accounts for 11 to 15 percent of the observed strength.

Residual Stress From Thermal Treatments

Fatigue specimens in which surface residual compressive stresses are developed by several common thermal treatments are perhaps the best available sources of data for evaluating surface weakness.

Nitriding and carburizing are among the most effective treatments for increasing the fatigue strength of steel. These treatments develop residual compressive stresses in the surfaces of case hardened parts. In Fig. 19 are shown the depths and magnitudes, as measured by the dissection method (Ref. 11), of the residual stresses developed in nitrided and carburized specimens by the expansion of the hard case during the hardening transformations.

Although the residual stresses from these treatments are less than was obtained by shot peening very hard steel, Fig. 2, they are more effective than the residual stress from shot peening would be when applied to the same steel without the hardened case. The greater surface yield strength, which is provided by the hardened case, prevents the loss of surface residual compressive stress during fatigue testing particularly in rotating beam, reversed bending, and reversed torsional tests. The surface protective stress in case hardened parts, therefore, remains fully effective under externally applied loads that would dissipate much or all of the protective stress induced by shot peening specimens having the same core hardness.

In reversed load testing, the manner in which residual stress is lost is shown in Figs. 20 (*A*) and (*B*), which represent a beam that is shot peened on both upper and lower surfaces. The magnitude of the peening stress on the respective surfaces is indicated by the distances OJ and $O'M$, each of which is equal to one half the elastic limit.

A downward acting load at O Fig. 20 (*A*), stresses the beam as indicated by the diagonal line N_1N_1. The line $J_2N_1N_1M'$ represents the sum of the residual stress and the externally applied bending stress. Since the stress represented by the distance Y_1J_2 exceeds the elastic limit of the metal, this portion of the residual stress on the upper surface is immediately lost, and only the portion N_1Y_1 remains. The fatigue strength of the specimen under repeated downward acting loads is not reduced by this loss, because it occurs on the side of the beam that is stressed only in compression. This loss, therefore, does not appreciably alter the tensile stress on the under surface, which determines the fatigue strength of the beam.

If the beam is now loaded from the opposite side by an upward acting load, Fig. 20 (*B*), a similar portion Y'_1M_2 of the residual compressive stress will be lost from the lower side of the specimen.

During the first downward acting load, Fig. 20 (*A*), the maximum tensile stress on the lower surface is represented by the distance $M'O'$. But when the load is reversed, Fig. 20 (*B*), the maximum tensile stress on the upper surface is represented by the greater distance K_2O. The increase in tensile stress is equal to the distance Y_1J_2 that was lost by the compressive yielding of this surface from the initial downward acting load.

After one complete reversed load cycle, the magnitude of the protective residual compressive stress on both surfaces is diminished, and the fatigue strength of the specimen may actually be reduced by the shot peening. The remaining protective stress may not be sufficient to offset the damage caused by the peening stress raisers. The same reduction of the residual compressive stress that is induced by low speed polishing of "ideal" rotating beam fatigue specimens also occurs. The polishing operation, however, does not add stress raisers, therefore any remaining stress is effective in overcoming surface weakness.

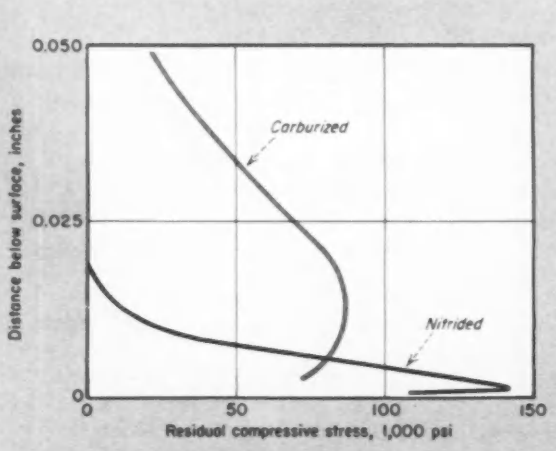


Fig. 19 Left — Residual stresses developed in nitrided and carburized specimens by the expansion of the hard case during the hardening transformations.

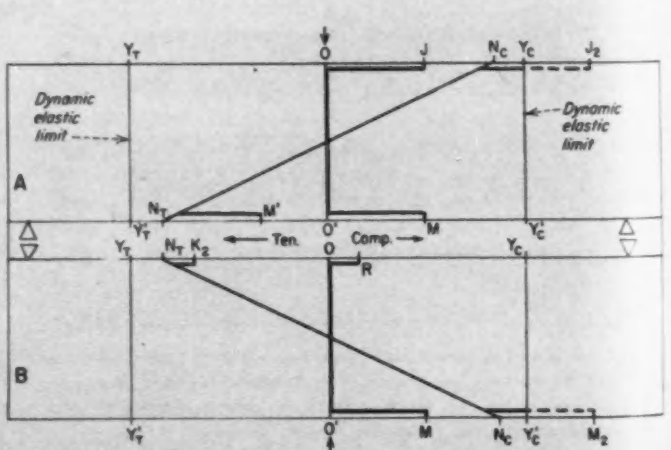


Fig. 20 Right — Manner in which residual stress is lost in a beam that is shot peened on both upper and lower surfaces. — (A) Downward acting load. — (B) Upward acting load.

Measurements of Residual Stress Loss

Buehler and Buchholtz (Ref. 20) reported experiments with annealed steel specimens in which residual stresses were developed by quenching in ice water from a tempering temperature of 1100 F. The yield stress of the steel is given as 44,200 psi. The residual stresses in one specimen were measured by the dissection method, which resulted in the internal stress pattern shown in Fig. 21 by the fine unbroken line labeled "Initial Residual Stress." The measured surface residual stress of 48,400 psi is not compatible with the estimated yield strength, but this discrepancy is not important to the results of the test as will appear.

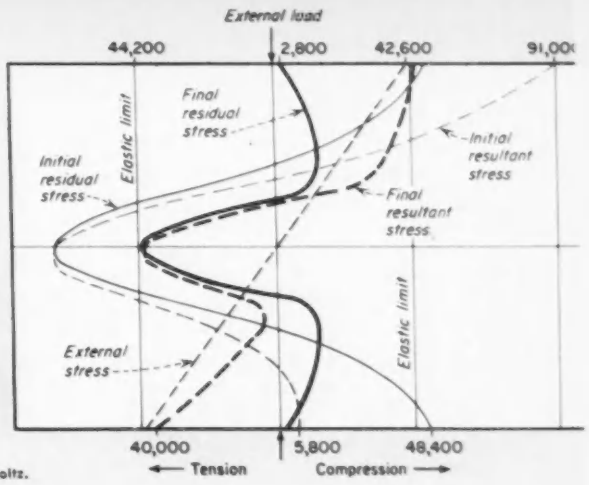


Fig. 21 — Residual stresses in annealed steel specimens that were quenched in ice water from a tempering temperature of 1100 F.

Replot from Buehler and Buchholtz.

A second specimen was dissected for stress measurements after it had been fatigue tested in reverse bending for 8,000,000 stress cycles at a nominal stress of 42,600 psi in either direction. The residual stress in this specimen is shown in Fig. 21 by the unbroken line labeled "Final Residual Stress." It will be seen that all of the surface residual stress except 2800 psi was lost during the test.

The reason for the loss is, of course, the same as was observed for the specimens Fig. 20. The sum of the initial surface residual stress and the surface stress from the external load is equal to 91,000 psi, as shown by the broken line labeled "Initial Resultant Stress." Since this stress exceeds the elastic limit of the steel by 47,000 psi, almost all of the residual stress was immediately dissipated.

The numerical values of stresses given in Fig. 21 cannot be expected to be of a high order of accuracy because, in addition to the usual errors, the results of the measurements from two specimens had to be used. However, the probable errors are small in comparison to the magnitude of the stress that was lost.

Residual Stress Retained by Case Hardened Specimens

The surface residual compressive stresses developed by case hardening processes such as nitriding, cyaniding, carburizing, and in some instances by induction and flame hardening, remain fully effective under external loads that completely dissipate the stresses induced by mechanical treatments such as polishing, shot peening, and rolling. The yield strength of the hardened case from each of the thermal treatments mentioned greatly exceeds the yield strength of the core of all ordinary fatigue specimens. For this reason, under the conditions of loading shown in Fig. 20, no loss of surface residual stress occurs in case hardened steel.

Since the residual compressive stress in the surfaces of nitrided specimens is great, and since this stress is not appreciably diminished by yielding, fatigue fractures originate in the non-nitrided sub-surface metal.

In non-nitrided specimens of identical sound steel, fatigue fractures invariably originate at the surface. Comparisons of the fatigue strength of nitrided and non-nitrided specimens, both of which are substantially alike in physical properties, therefore, may be used to indicate the magnitude of surface weakness. Since in both kinds of specimens the fracture originates in substantially identical material, it is only necessary to know the actual stress at the point of fracture origin in the nitrided and in the non-nitrided specimens.

The fatigue tests should be made under load conditions that will cause failure of both kinds of specimens in approximately the same number of stress applications or the endurance limit for both types should be established.

Surface Weakness Estimated From Nitrided Specimens

Dissection stress measurements from identical non-nitrided and nitrided specimens that have been fatigue tested to failure are not available. Rough estimates of the relative strength of surface and subsurface steel, however, may be made from published fatigue data obtained from suitable specimens. Among the test data that may be used for this purpose are the results of tests reported by Johnson and Oberg (Ref. 21) of Wright Field. This report is selected principally because of the availability of the photographs Fig. 22, which show the sub-surface origin of the failure in the nitrided specimens as compared to the surface origin of the fractures in the non-nitrided specimens.

The relative fatigue strength of the two groups of rotating beam specimens is shown in the SN chart

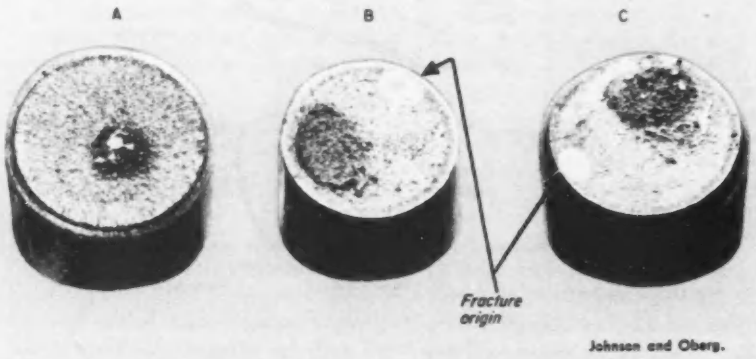


Fig. 22—Origin of failures in Johnson and Oberg specimens. Fatigue fracture in nitrided specimens B and C originated in core. Fatigue fracture in non-nitrided specimen A originated at surface.

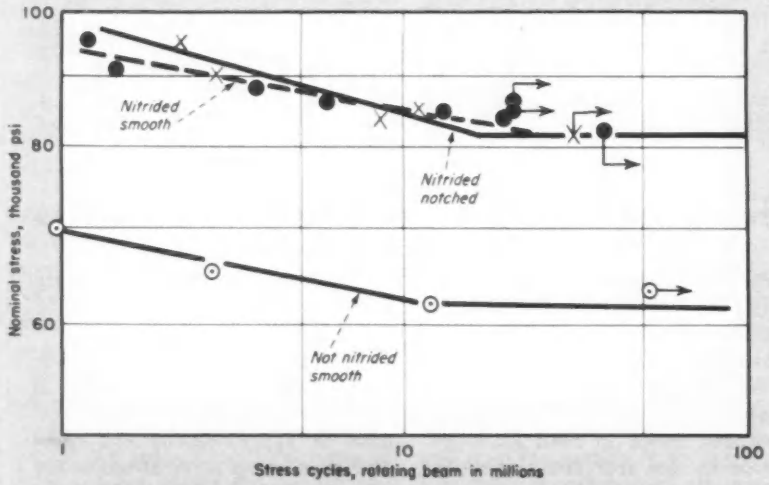


Fig. 23—Estimated relative fatigue strength of the nitrided and non-nitrided rotating beam specimens of Johnson and Oberg.

Replot from J. B. Johnson and T. T. Oberg, *Metals and Alloys*, Vol. 5, p 130, 1934.

Fig. 23, which is a replot of the original data. From these graphs, it is seen that the nominal fatigue endurance limit of the non-nitrided specimens, in which the fracture originated at the surface, is 62,000 psi as compared to the nominal endurance limit of 82,000 psi for the nitrided specimens, in which the fracture originated in the sub-surface metal. The reported stresses are presumably the calculated values at the surfaces of the 0.3 in. dia specimens. The actual stresses are not known for either kind of specimen because of the presence of residual stresses in both types.

In the non-nitrided specimens, the stress causing fractures was probably less than the value shown in the chart because, in the process of polishing, residual compressive stresses are induced in the surface by the polishing. The effective stress at the surface of these specimens is, therefore, the stress recorded in the chart minus the retained residual compressive stress from the polishing. Polishing stress measurements indicate that residual compressive stress from this source may exceed 20,000 psi (Ref. 11). But if compressive stress of this magnitude was induced during polishing, as indicated in Fig. 20, it is possible that a part was dissipated by yielding during the reversed bending fatigue tests.

Probably the best estimate of the discount that should be applied to the fatigue strength of the Johnson and Oberg non-nitrided specimens, because of the residual stress from polishing, are the data from the H. F. Moore specimens given in Table I. If eleven percent of the observed 62,000 psi fatigue strength is attributed to polishing residual stress, the actual fatigue strength of the polished surfaces was approximately 56,000 pounds per square inch.

Reconstructed Stress in Nitrided Specimens

The actual stress in the sub-surface metal of nitrided specimens, at the points of origin of the fractures, would be greater than was indicated for this portion of the beam by the calculations. To convey an idea of the stress pattern in the nitrided specimens shown in Fig. 22, the diagram Fig. 24 has been constructed from Johnson and Oberg test data.

This diagram assumes that the magnitude and depth of the measured residual stress shown in Fig. 19 applied also to the 0.3 in. specimens used by Johnson and Oberg. The axial compressive force from the residual stress in the nitrided case of this composite specimen must be balanced by a tensile force of equal magnitude acting on the core. Since the area of the core is greater than the area of the case, the maximum residual tensile stress in the core would be approximately

one-tenth of the residual compressive stress in the case.

Measurements of the photographs, Fig. 22, indicate that the failures originated approximately at 0.025 to 0.030 in. below the surface, which serves to indicate the depth of the maximum resultant tensile stress. It is seen that the sum of the stress from the external load and the residual tensile stress in the core at the point of maximum stress is 78,000 psi, which is 95 percent of the nominal surface stress. By comparison, the stress as calculated from the nominal stress gradient at a depth of 0.027 in. is 82 percent of the surface stress. However, the stress as indicated in the SN diagram, Fig. 23, is presumably the nominal surface stress and not the nominal stress at the depth of the fracture origin.

Because of the many uncertainties in the foregoing estimates of the actual stress, the present purpose will be better served by using the reported nominal test stresses for comparing the relative fatigue strength of the surface and sub-surface metal. From the SN chart, Fig. 23, the ratio of surface strength to sub-surface strength at the endurance limit is 62,000 to 82,000 psi, that is, based on the uncorrected stresses the fatigue strength of the surface is about 75 percent of the fatigue strength of submerged metal.

Nitrided Surface Stresses Always Compressive

The maximum nominal stress that was applied to these nitrided specimens during the reversed bending test, it is interesting to note, was less than 100,000 psi. Since the residual compressive stress in a nitrided case, as shown in Fig. 19, approximates 140,000 psi, it is apparent that the stress from the reversed bending load in much of the nitrided case was not tension at any time. This deduction accounts for the fact that the hard surfaces of case hardened specimens are not "brittle," and that Johnson and Oberg found that sharp notches could be ground through 70 percent of the nitrided case before the fatigue strength of the specimen was adversely affected. Fatigue fractures can only develop and propagate in regions that are stressed in tension.

Incidentally, the maximum compressive stress in the case was, of course, equal to the sum of the residual compressive stress and the stress from the test load, which on the basis of the nitriding stress shown in Fig. 19 approximated 230,000 psi, as shown in Fig. 23 near the upper surface.

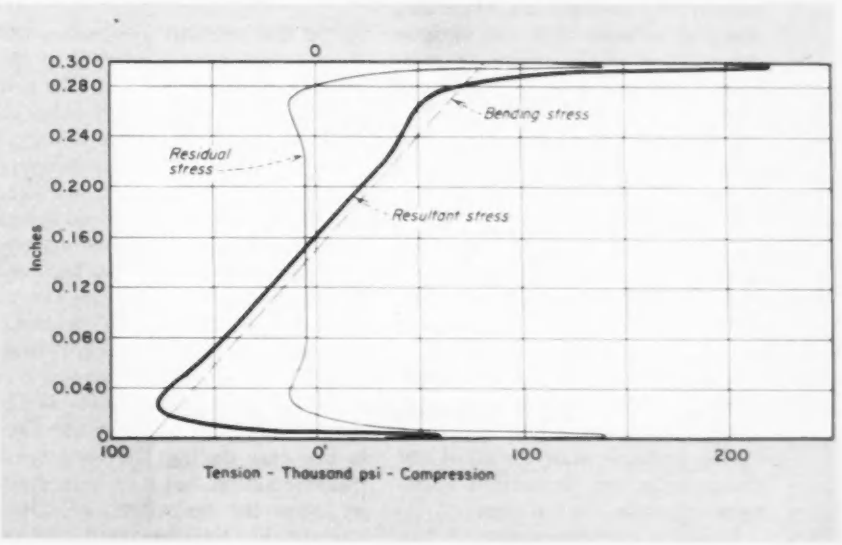
Other reports relating to fatigue strength comparisons of nitrided and non-nitrided specimens are suitable for indicating the relative strength of surface and sub-surface metal. In all the references, which will be cited, care was used to develop the same physical properties in the non-nitrided steel of both kinds of specimens. The fatigue strength of the surface will be given in percent of the core fatigue strength based upon the uncorrected stress values given in the several reports.

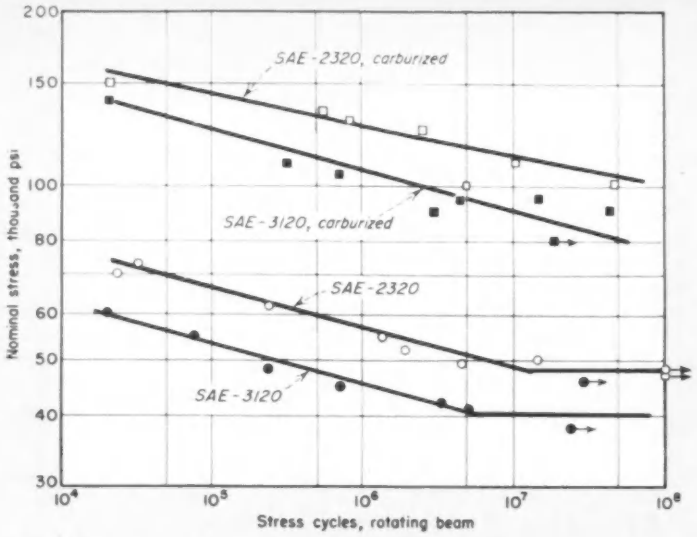
Von O. Hengstenberg and Mailaender (Ref. 22) tested three groups of specimens made from different kinds of steel. Subsurface fractures were reported to have occurred in the nitrided specimens. The surface fatigue strength is shown as 69 percent, 73 percent, and 78 percent of the fatigue strength of sub-surface metal for the respective steels.

Mailaender (Ref. 23) reported fatigue tests that indicate the fatigue strength of the surface to be 71 percent of the fatigue strength of sub-surface metal.

Sutton (Ref. 24) compared nitrided and non-nitrided specimens by repeated axial loads as well as by rotating beam loaded in bending. This comparison is an interesting variation because, as is evident from inspection of Fig. 24, the tensile stress in the nitrided specimens is proportionately greater under axial loads than under bending loads. Sutton's uncorrected stress data give the surface fatigue strength of rotating beam specimens as 82 percent, and the axial stressed specimens as 97 percent, of the fatigue strength of sub-surface metal. Presumably these values would approach one another, and they would also become smaller, if corrected stresses were available.

Fig. 24 — Constructed stress pattern in the nitrided specimens shown in Fig. 22 used by Johnson and Oberg.





Replot from H. F. Moore and N. J. Alleman; A.S.S.T. Trans., Vol. 13, 1928.

Fig. 25 — Fatigue strength of carburized and non-carburized SAE 3120 and SAE 2320 steel rotating beam specimens.

Surface Weakness Estimated From Carburized Specimens

Tests that have been made to measure the relative fatigue strength of carburized specimens can also be used for estimating surface weakness. Among the fatigue data, which are suitable for this purpose, are the results of two series of tests conducted by H. F. Moore and N. J. Alleman (Ref. 25) on carburized and non-carburized SAE 3120 and SAE 2320 steel.

These data are shown in the replotted SN chart Fig. 25. From this chart it is seen that the fatigue strength of the surfaces of non-carburized specimens is, in terms of uncorrected stress, only one-half as great as the fatigue strength of the equivalent material in the core of the carburized specimens.

The data reported by Moore and Alleman do not mention sub-surface fractures. But properly performed carburizing and quenching develop residual compressive stress in the hardened layer somewhat as is indicated by the measured stress shown in Fig. 19. Also, since the diameter of the specimens was the standard 0.3 in., probably many or all of the fractures in the carburized specimens originated in the core.

Through correspondence, it has been learned that Professors Moore and Alleman recalled that many sub-surface fractures did occur. At the time when these tests were made the effects of residual stresses were not appreciated, and since the point of origin of the fractures was not important to the purpose of the tests, no record was made of their frequency.

Only Nominal Stress Available

As in the nitrided specimens, the stresses that are recorded in the SN chart, Fig. 25, are nominal only and the actual stresses at the point of fracture origin can only be approximated by dissection measurements of the residual stresses in the carburized and the uncarburized specimens. Polishing stresses were probably induced and may have been retained on the surfaces of the uncarburized specimens, in which event the stress for these specimens will have to be revised downward.

Carburizing would have developed residual compressive stresses in the case during the hardening transformation, but it is hazardous to judge the magnitude of these stresses by the measured values shown in Fig. 19. These stress measurements were made on a specimen carburized to a depth of approximately 0.055 in., whereas, the carburizing depth of the Moore and Alleman specimens is given as 0.025 in. for the SAE 2320 steel and 0.032 in. for the SAE 3120 steel.

Both series of specimens were quenched in oil directly from the carburizing box without subsequent treatment. The surface hardness is recorded as Rockwell C 64-65. The tests are disappointing in that the non-carburized specimens were tested in the as rolled condition, therefore, the physical properties of the surface of these specimens may differ from the physical properties of the core of the carburized specimens.

Surface Only one Half as Strong as Sub-Surface

There can be no doubt that the cores of the carburized specimens were residually stressed in tension. It is, therefore, reasonable to assume that the actual differences in the stresses at the points of origin of the fractures were greater than the difference in the nominal stresses shown in the SN chart. In the absence of more precise data, the comparisons will be made on the uncorrected stresses as reported, from which it is found that the surface fatigue strength for both steels is approximately 50 percent of the strength of the sub-surface metal.

Although the Moore and Alleman report does not recognize residual stresses, it does comment on the test results as follows: "... where the core and case meet, it will be found . . . that at the endurance limit (the) computed stress is . . . usually 50 percent higher than the endurance limit of the virgin metal. This would indicate that at the outer edge of the core the strength of the material is at least as great as that of the virgin metal."

Woodvine (Ref. 26) fatigue tested three kinds of steel in the carburized and non-carburized condition. In preparing the specimens, Woodvine reports that the non-carburized specimens were given the same heat-treatment as the carburizing specimens, except that they were heated in sand instead of in a carburizing environment. Sub-surface fractures were found in the carbur-

ized specimens. The surface fatigue strengths of the three steels based on uncorrected stresses as reported were 51, 55, and 62 percent of the strengths of the sub-surface steel in their respective groups.

CYANIDING DEVELOPS RESIDUAL STRESS. Residual stresses that are developed by cyaniding have not yet been measured by the General Motors Research Laboratory, but the measurements made by Becker and Phillips (Ref. 27) will serve as qualitative if not quantitative evidence. The residual stresses in the hardened case are shown to be compressive.

The fatigue strength of cyanided and non-cyanided specimens of 0.16 carbon steel as reported by Lea (Ref. 28), indicate that in terms of uncorrected stresses, the non-cyanided surface of his test specimens was 55 percent as strong as the non-cyanided sub-surface steel.

Ductile and Brittle Materials

The ratio of surface fatigue strength to sub-surface fatigue strength is probably not the same for all structural materials. It is also probable that in any material this ratio varies with such properties as hardness and ductility.

The surface of a specimen of very hard steel, in which the hardness is uniform throughout, does not have the surface protection of residual compressive stresses that occurs in case hardened surfaces. Through-hardened steel specimens will therefore be more susceptible to surface fatigue failure, because the low ductility of the hard metal reduces the ability of the surface to relieve local tensile stresses by plastic adjustment. Such specimens are said to be "notch sensitive" or "brittle."

Effect of Surface Stress Raisers in Statically Loaded Ductile Metals

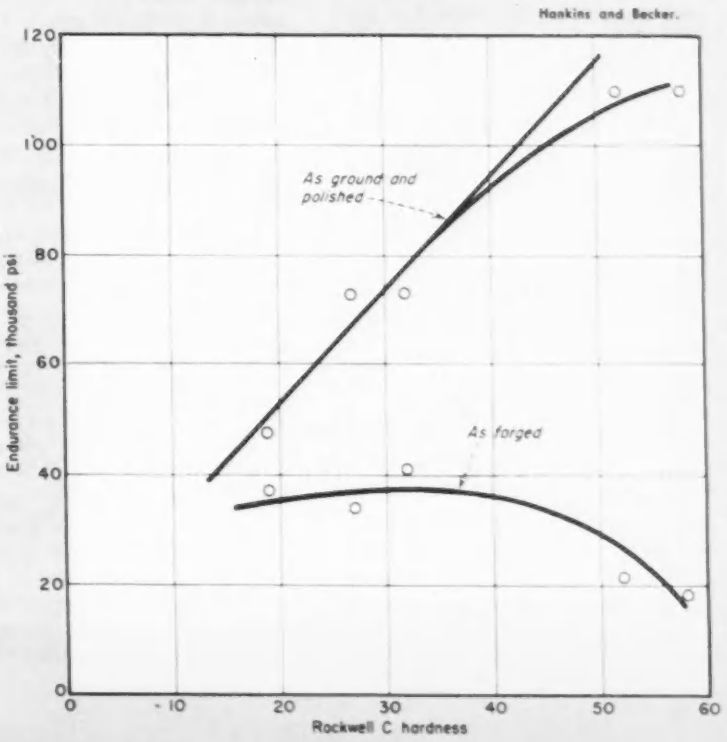
In ductile metals under static loading, the effect of surface stress raisers is negligible. Any local increase of stress in a stress raiser, for example a scratch, is dissipated by plastic flow of the metal affected by the stress raiser. Such localized plastic adjustments may be said to occur on a micro scale. The result is that the surface stress becomes quite uniform in spite of surface imperfections. Also in statically loaded ductile specimens, the effect of the inherent weakness of surfaces is also reduced by general plastic yielding of the surface at nominal stresses less than the yield strength of the core.

Plastic flow of surface metal has been shown by Norton (Ref. 29), who measured the residual stress in a tensile specimen after it had been statically loaded slightly above the nominal elastic limit. He found that a surface layer of measurable depth was residually stressed in compression, presumably because of the greater plastic extension of the surface than of the sub-surface metal. Such surface extension probably occurs, but on a greatly reduced scale, when ductile metals are subjected to repeated loads at lower stresses, as in fatigue tests. Fatigue fractures therefore are brittle in appearance in metals that will show extensive necking under static tests.

Plastic relief of local stresses becomes increasingly difficult as ductility is reduced until, as in a material such as glass, local stress raisers and general surface weakness are equally effective whether the specimen is loaded statically or by repeated loads. For this reason, the strength of glass under repeated loads is identical with its static strength. In terms of the relative magnitudes of static and fatigue strength of metals, this fact means that because of surface weakness the static strength of glass is reduced to equal its strength under repeated loads.

A suggestion of this increased surface vulnerability with decreased ductility is seen in the experiments of Hankins and Becker (Ref. 30). As shown in the plot of the data they obtained from ground and polished steel specimens, Fig. 26, the endurance limit increases linearly with hardness until Rockwell C 40 is reached. At higher hardness levels, the reduced ductility does not permit adequate plastic relief of the tensile stress in local stress raisers. The rate of fatigue strength increase is therefore sharply reduced.

Fig. 26—Increase of surface vulnerability with hardness in ground and polished steel specimens.



although the strength of the sub-surface steel presumably continues to increase at high hardness levels.

Improving Static and Impact Strength

Surface weakness affects the strength of materials under load conditions other than repeated stresses.

Materials of low ductility fail at low stress by brittle fractures under static and impact loads, because they are not capable of yielding to relieve local high stresses whether from surface imperfections, residual micro stresses or just plain undefined surface weakness. Brittle materials will therefore suffer from surface imperfections whether the applied load is static, impact, or repeated fatigue loads. The imperfections may or may not be in the form of recognizable stress raisers but, regardless of smoothness, the surfaces are weaker than sub-surface material.

Since completely brittle materials fail only by tensile stresses (Ref. 31) their strength, under various kinds of loading, is increased by inducing residual compressive stresses in the fracture sensitive surface layer. Steel having the hardness of the case of carburized and nitrided specimens would be notch sensitive and therefore brittle, except as shown in Fig. 19 that the hard layers are residually stressed in compression.

Prestressing by mechanical means can only be accomplished in materials having some ductility. Fig. 2 shows that steel of Rockwell C 64 hardness is sufficiently ductile to respond to shot peening. The peening, which was not of high intensity, is seen to have developed a residual compressive stress that extended to a depth of 0.009 in. and a magnitude of 150,000 psi. Mechanical prestressing can, therefore, be expected to increase the static, the impact, and the fatigue strength of many hard metals.

Mechanical prestressing should also be effective in reducing the "brittleness" acquired by hardened steel that is uniformly hard throughout. Such prestressing should also reduce the hazard of fractures that often occur because of reduced ductility at low temperature. In fact, the effectiveness of surface residual compressive stress should increase as the "brittleness" increases.

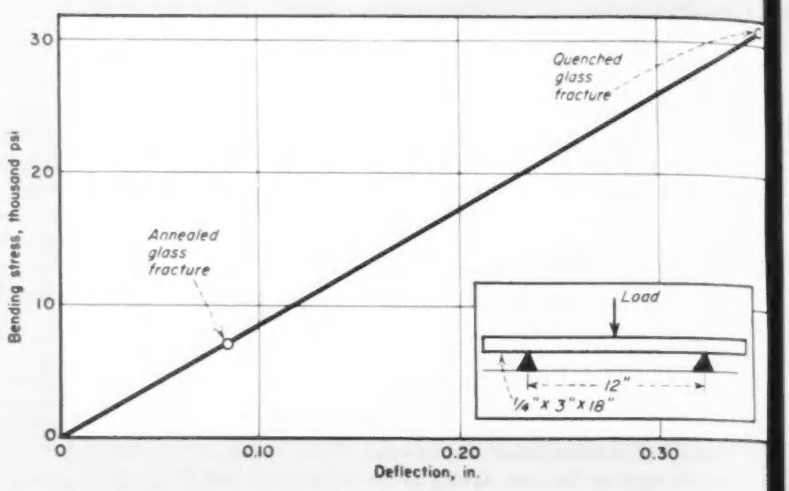


Fig. 27 — Stress-strain curves in annealed and in quenched glass. Insert shows specimen dimensions and loading method.

Strength Increases As Ductility Decreases

Materials, such as glass, in which measurable plastic flow cannot occur at ordinary temperatures may have their static strength and impact strength increased by an amount equal to the magnitude of residual compressive stress that can be developed in the surface of the specimen. Thus the static, impact, and fatigue strength of glass (Ref. 32) can be increased several hundred percent by quenching from a temperature at which the glass is plastic.

When a glass plate or rod is quenched from the plastic range, the surfaces cool and contract while the core deforms plastically to conform to the altered surface dimensions. Subsequent cooling and contracting of the core develop a tensile stress in the sub-surface glass and a corresponding compressive stress in the surfaces. The residual surface stress that is trapped by this process is three or more times as great as the tensile strength of ordinary annealed glass.

The increase in static strength obtained by prestressing is shown by the experiment recorded in Fig. 27. Static bending loads were applied on annealed and on quenched glass beams as shown at the lower right of the chart. The annealed glass failed at a stress of 7,000 psi, whereas the prestressed glass failed at a nominal stress of 32,000 pounds per square inch.

Residual Stress Measurements in Glass

Residual stresses in quenched glass have been measured by Littleton (Ref. 32), by means of photo-elastic strain patterns. He found that the static strength of quenched glass exceeds that of annealed glass by an amount equal to the magnitude of the residual compressive stress that is developed during quenching. This observation means that the static strength of the quenched glass beam, Fig. 27, was increased from a nominal stress of 7,000 psi to nominal stress of 32,000 psi by a surface residual compressive stress of 25,000 psi.

Based upon the residual stress experiments of Littleton and the static strength comparisons of Fig. 27, the stress diagrams of the latter specimens are reconstructed in Fig. 28. Since the annealed glass plate was presumably free from residual stresses of significant magnitude, the stress in this specimen at the instant of fracture is repre-

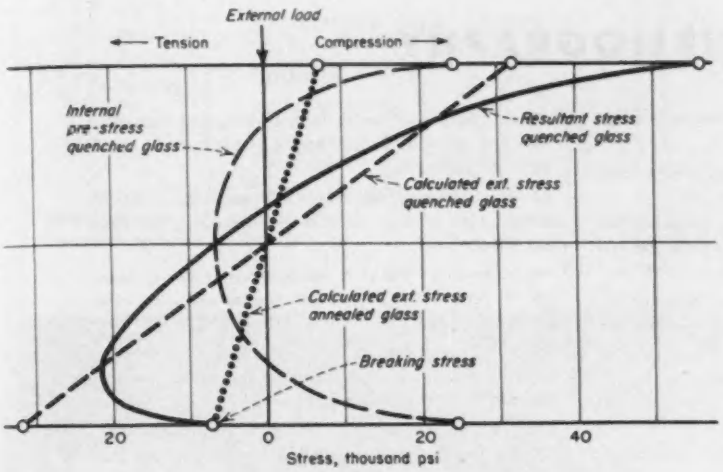


Fig. 28—Reconstructed diagrams to show stress distribution in glass beam specimens when they were loaded to failure.

sented by the diagonal dotted line labeled "Calculated Ext. Stress Annealed Glass." This line is drawn to conform to the static strength test, by which the fracture was found to occur on the lower surface at 7,000 psi tensile stress.

The nominal stress from the external load on the quenched glass at the instant of fracture is represented by the dashed diagonal line labeled "Calculated Ext. Stress Quenched Glass." The surface stresses are shown as 32,000 psi to conform with the Fig. 27 nominal test stress.

The distribution of the residual stress that was developed by the quenching operation is shown by the symmetrical curved line labeled "Internal Prestress Quenched Glass," which is proportioned in accordance with Littleton's photoelastic measurements. The residual compressive stress on both surfaces is shown as 25,000 psi for the reason stated previously.

The resultant stress shown in Fig. 28 is the algebraic sum of the residual stress and the nominal stress from the external load. It is represented by the unbroken curved line labeled "Resultant Stress Quenched Glass," which extends from 57,000 psi compression to 7,000 psi tension. It will be seen that the maximum tensile stress occurs in sub-surface glass, and reaches a value of approximately 21,000 psi but, because of surface weakness, the fracture presumably originated at the surface from a tensile stress of only 7,000 pounds per square inch.

Strength Increased Four Hundred Percent

Although the quenched glass beam, Fig. 27, was more than 400 percent stronger than the annealed glass beam, the surface fracture strength was the same for both specimens. By these tests, the surface of polished plate glass is found to be less than 33 percent as strong as subsurface glass. Many years ago Littleton said, "We never test the strength of glass; all we test is the weakness of its surface." It now appears that this statement applies equally well to repeatedly loaded metals and perhaps to many other materials.

Available test data (Ref. 33) show that the fatigue strength of glass subjected to repeated bending loads is substantially the same as its static strength. Its impact strength may be somewhat greater because of the short duration of loading. Since there is no change in stress by plastic flow, the strength of glass is not affected by the manner of loading, except for the well-known loss of strength (Ref. 31-34) as the duration of loading is increased.

It is to be expected that the behavior of metals will approach the behavior of glass as their ductility is decreased. Very hard steel tested at a very low temperature should be relatively insensitive to the manner of loading, and surface weakness should increase. Its fatigue strength should approach its static and impact strengths, and surface prestressing should remain effective regardless of loading method.

EDITOR'S NOTE: In the near future PRODUCT ENGINEERING will publish a sequel article "Fatigue Failures Are Tensile Failures" by J. O. Almen, in which he will discuss how unexpected improvement in fatigue strength is often obtained from processes that are applied for other purposes.

ACKNOWLEDGMENTS

Valuable assistance was given by many co-workers in the General Motors Corporation in the preparation of this article. Particularly appreciated is the help given by: William E. Miller and Carmin Guerrera, who performed the delicate dissection operations and the precise measurements by which the residual stresses in many specimens were reconstructed. Mary Elizabeth Kerr performed the calculations and prepared the many charts that are used as illustrations. Robert Schilling, R. L. Mattson, and Maurice Olley offered many helpful criticisms and suggestions.

BIBLIOGRAPHY

1. Unpublished data General Motors Research Laboratories.

2. J. O. Almen, Shot Blasting to Increase Fatigue Resistance, *S. A. E. Journal*, v. 51, No. 7, 1943, p. 248.

3. A. Thum and W. Bautz, Ursachen der Steigerung der Dauerhaltbarkeit gedruckter Staebe, *Forschung des Ingenieurwesens*, v. 6, No. 3, May/June, 1935, p. 121-8.

4. O. Foeppl and A. Thum, Steigerung der Dauerhaltbarkeit durch Oberflaechen-druecken, *Zeitschrift des Vereines Deutscher Ingenieure*, v. 77, No. 50, 1933, pp. 1335-1337.

5. O. J. Horger and H. R. Neifert, Improving Fatigue Resistance by Shot Peening, *Experimental Stress Analysis*, v. II, No. 1, May, 1944, p. 178.

6. O. J. Horger and C. H. Lipson, Automotive Rear Axles and Means of Improving their Fatigue Resistance, *A.S.T.M. Technical Publication*, No. 72, presented June, 1946.

7. R. L. Mattson and J. O. Almen, Effect of Shot Peening on the Physical Properties of Steel, Final Report, Part II, O.S.R.D. No. 4825, published by National Defense Committee of the Office of Scientific Research and Development, War Metallurgy Division, 1945, pp. 65-66.

8. D. K. Crampton, Effects of Internal Stresses on Properties of Drawn Brass Tubes, *Trans. Amer. Mining and Metallurgical Engineers*, v. 104, 1933, p. 210.

9. R. L. Mattson and J. O. Almen, Effect of Shot Peening on the Physical Properties of Steel, Final Report, Part II, O.S.R.D. No. 4825, published by National Defense Committee of the Office of Scientific Research and Development, War Metallurgy Division, 1945, pp. 98-99.

10. E. Heyn, Internal Strain in Cold Wrought Metals and Some Troubles Caused Thereby, *Journal of the Inst. of Metals*, v. 12, 1914, p. 137.

11. J. O. Almen, Peened Surfaces Improve Endurance of Machine Parts, *Metal Progress*, Feb., 1943, p. 215.

12. H. H. Miller, New Cut Wire Shot Big Boon to Peening, *S.A.E. Journal*, v. 57, No. 8, Aug., 1949, p.44.

13. R. L. Mattson and J. O. Almen, Effect of Shot Peening on the Physical Properties of Steel, Final Report, Part 1, O.S.R.D. No. 3274, published by the National Defense Committee of the Office of Scientific Research and Development, War Metallurgy Division, 1944, p. 81.

14. R. L. Mattson and J. O. Almen, Effect of Shot Peening on the Physical Properties of Steel, Final Report, Part 1, O.S.R.D. No. 3274, published by the National Defense Committee of the Office of Scientific Research and Development, War Metallurgy Division, 1944, pp. 56 and 57.

15. R. L. Mattson and J. O. Almen, Effect of Shot Peening on the Physical Properties of Steel, Final Report, Part II, O.S.R.D. No. 4825, published by National Defense Committee of the Office of Scientific Research and Development, War Metallurgy Division, 1945, pp. 43 and 44.

16. R. L. Mattson and J. O. Almen, Effect of Shot Peening on the Physical Properties of Steel, Final Report, Part III, O.S.R.D. No. 6647, published by National Defense Committee of the Office of Scientific Research and Development, War Metallurgy Division, 1946, p. 110.

17. O. J. Horger, Increasing Fatigue Strength of Press Fitted Axle Assemblies by Surface Rolling, A.S.M.E. Trans., v. 58, 1936, pp. A91-A98.

18. H. F. Moore, Size Effect and Notch Sensitivity in Fatigue Tests of Steel, *A.S.T.M. Proceedings*, v. 45, 1945, pp. 516-519.

19. D. Markoven and H. F. Moore, Effect of Size of Specimen on Fatigue Strength, *A.S.T.M. Proceedings*, v. 44, 1944, pp. 143 and 144.

20. H. Buehler and H. Buchholtz, Die Wirkung von Eigenspannungen aus die Biegeschwingungsfestigkeit, *Stahl u. Eisen*, v. 53, 1933, pp. 1330-1331.

21. J. B. Johnson and T. T. Oberg, Effect of Notches on Nitrided Steel, *Metals and Alloys*, v. 5, 1934, pp. 129 and 130.

22. O. Hengstenberg and R. Mailaender, Bending Fatigue of Nitrided Steels, *Kruppsche-Monatschefts*, v. 11, Sept.-Oct. 1930, p. 252.

23. R. Mailaender, Ueber die Dauerfestigkeit von nitrierten Proben, *Tech. Mitt. Krupp*, Hft 2, July, 1933, pp. 53-58, also *Z. Ver. deut. Ing.*, v. 77, 1933, pp. 271-274.

24. H. Sutton, Fatigue Properties of Nitrided Steel, *Metal Treatment*, v. 2, 1936, pp. 89-92.

25. H. F. Moore and N. J. Alleman, Fatigue Tests of Carburized Steel, *Trans. Amer. Soc. Steel Treat.*, v. 13, 1928, pp. 405-419.

26. J. G. R. Woodvine, Behavior of Case-Hardened Parts Under Fatigue Stresses, Carnegie Schol. Mem., *Iron Steel Inst.*, v. 13, 1924, pp. 197-237.

27. M. L. Becker and C. E. Phillips, Internal Stresses and Their Effect on the Fatigue Resistance of Spring Steel, *The Jour. of Iron and Steel Inst.*, v. 133, No. 1, 1936, pp. 427-442.

28. F. C. Lea, Effect of Discontinuities and Surface Conditions on Failure Under Repeated Stress, *Engineering*, v. 144, 1937, pp. 87-90, 140-144.

29. J. T. Norton, X-ray Methods in the Field of Residual Stresses, *Experimental Stress Analysis*, v. 11, No. 1, pp. 157-160.

30. G. A. Hankins and M. L. Becker, The Fatigue Resistance of Unmachined Forged Steels, *The Jour of the Iron and Steel Inst.*, v. 127, No. 2, 1932, p. 205.

31. F. W. Preston, The Mechanical Properties of Glass, *Journal of Applied Physics*, v. 13, Oct., 1942, pp. 623-634.

32. J. L. Littleton, Jr., A New Method for Measuring the Tensile Strength of Glass, *Physical Review*, v. 22, 1923, pp. 510-516.

33. Unpublished data General Motors Research Laboratories.

34. A. A. Griffith, The Phenomena of Rupture and Flow in Solids, *Transactions Royal Society*, A, v. 221, 1920, p. 162.

35. Hayo Foppl, Messung der Spannungen und der Plastischen Verformungen an Oberflachengedruckten Probekorpern, Mitteilungen des Wohler-*Institute Braunschweig*, Heft 41, Friedr. Vieweg & Sohn, Braunschweig, 1948.

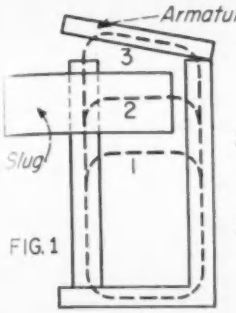


FIG. 1

1 Magnetic flux path immediately after coil circuit closure

2 Magnetic flux path a moment later

3 Final flux links armature to cause operation

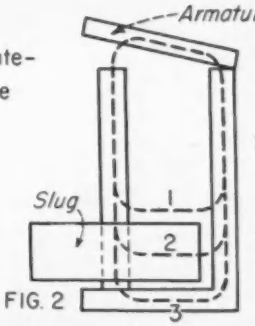


FIG. 2

1 Magnetic flux path immediately after coil circuit closure

2 Magnetic flux path a moment later

3 Final flux links slug. Slug will now be effective on release time

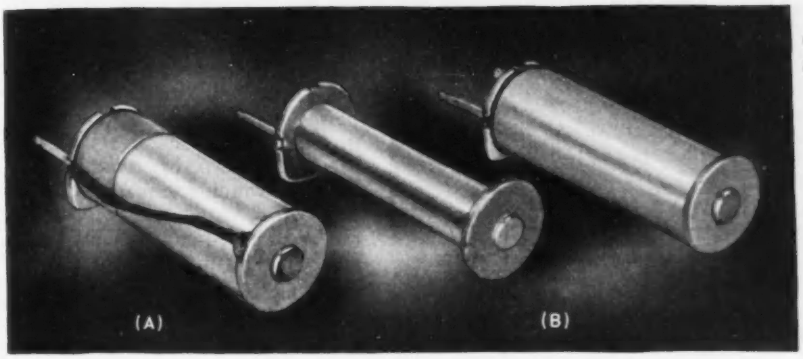


FIG. 3

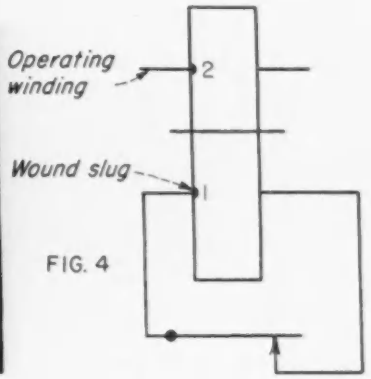


FIG. 4

Fig. 1—Rise of flux in magnetic circuit of a relay having a coil equipped with an armature end slug.
Fig. 2—Slug on heel end or end of spool farthest from the armature provides only release time delay.
Fig. 3—(A) Heel end slug coil, which is relatively heavy. (B) Sleeve coil spool and wound coil.
Fig. 4—Short circuited front end section winding gives slugging effect to produce operate time delay.

Telephone Type Relays

Coil Constructions and Timing

Selection of coil to obtain time required for relay operation, holding, and release demanded by the circuit control application. Performance and life characteristics of typical coil constructions and stepping switches are compared.

V. E. JAMES
Staff Engineer, Industrial Control Division
Automatic Electric Company

GENERAL RELAY FUNDAMENTALS were reviewed, overall performance advantages and limitations of telephone type and power type relays were discussed, and some economic factors were studied in Part I of this article, PRODUCT ENGINEERING, October, p 130. But certain characteristics need further examination before the designer arrives at a satisfactory choice of a particular construction of telephone type relay for a specific purpose.

So many factors affect relay timing, especially for combinations of operate and release timing, and the possibilities are so varied and interlinked with so many different variable factors, that it is possible here to cover the subject only in an incomplete manner.

The ordinary or unslugged single winding coil of a telephone type relay provides "fast operation and fast release". Even so, various factors affect the operating speed of a "fast coil" relay. Coils with a large number of turns of wire have a large amount of self-inductance, other factors being the same, and hence the build-up of magnetic flux in their cores is retarded more than in coils having fewer turns of wire. This factor should not be ignored when selecting coils for a fast operate time.

Self-inductance also varies with the amount of iron in the magnetic circuit; hence, miniature relays are inherently faster to operate than those using heavier masses of iron for the magnetic path.

Since the ratio of inductance to resistance, or L/R, of the relay coil is

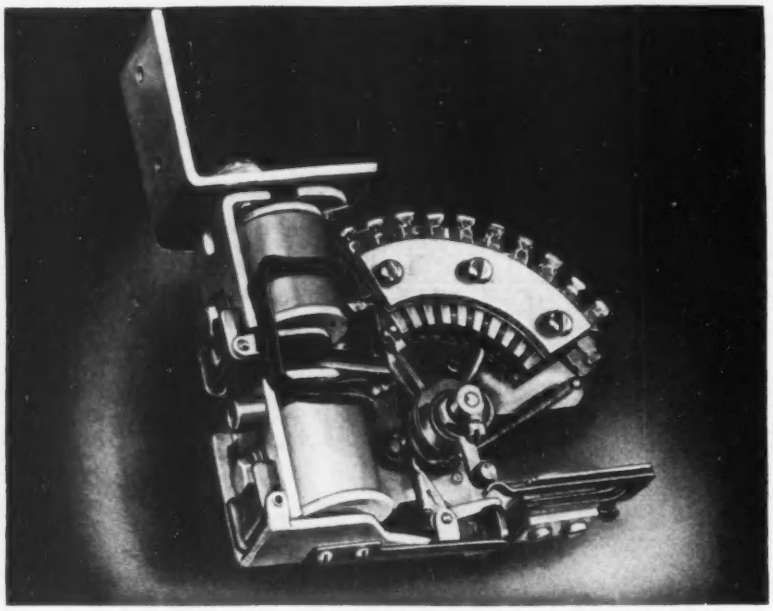


Fig. 5—A direct drive telephone type single motion stepping switch that returns over same path from contact to which it was stepped. Action varies with power input.

the "time factor", a provision of excessive "steady state" power in the relay coil is to be avoided. Where speed of operation is essential, maximum speed of operation will be obtained by adjusting the wire size and turns relationship so that the power provided to operate the relay is not too greatly in excess of the minimum acceptable value, by keeping the coil resistance high, and by keeping the number of turns of wire as low in number as practicable.

Many rules of thumb can be employed as a guide, but the usual practice is to use as fine a wire as possible when supplying a total steady state value of ampere-turns of 2 to 3 times that which barely provides operation. This practice is tempered by the relay engineer to fit the voltage extremes to be encountered and the spring loading flexibility.

In telephone relays, it is easier to provide fast operation on voltages from 100 to 200 volts than for those in only a fraction of this range, other factors being the same. The spring loading also affects the operating speed of the relay, since operation is not obtained until the magnetic flux has had time to build up to the required value, and this time varies with the mechanical loading. Fast release, therefore, should be aided in part by heavy mechanical loading.

Another factor, usually more important in securing fast release, is the size of the residual gap between armature and coil core in the fully operated position. Since an increase in the gap adds to the armature stroke, it also has a noticeable effect on operating speeds, and one that sometimes cannot be ignored or totally offset by compensating factors. The amount of power provided as compared to the minimum barely required for operation is also a factor in speed of release. Since some of these factors that favor speedy operation also hinder speedy release and vice versa, a careful balancing of all these factors (and some others as well) by the relay engineer is necessary in critical timing applications.

Table I—Timing Limits of Telephone Type Relays

Slug or Sleeve Position	Large Size		Miniature Size	
	Operate Time, milli-sec	Release Time, milli-sec	Operate Time, milli-sec	Release Time, milli-sec
Arm. End	100 to 150, max 25 to 40, min	300 to 750, max 75 to 200, min	25 to 40, max 10 to 15, min	40 to 120, max 15 to 25, min
Heel end	4 to 25	300 to 750, max 75 to 200, min	7 to 10	40 to 120, max 15 to 25, min
Sleeve (full length of coil core)	25 to 75	300 to 750, max 75 to 200, min	10 to 25	40 to 120, max 15 to 25, min

MULTI-WOUND COILS. Multi-wound coils are designed for many purposes. One design employs one winding for operation of the relay and another for holding the relay operated after the first winding has been de-energized. Sometimes the windings are balanced so that with both energized in magnetic opposition no operation takes place, but with either winding de-energized operation of the relay results. The reverse situation is the "buck down" usage, where the relay operates on one winding and is magnetically "starved" down by energization of the other winding in magnetic opposition. Also, more than two windings can be provided and combined to obtain variations in operation.

DELAYED ACTION COILS. When the core of a relay coil is encircled by a solid ring of electrically conducted material, called a slug or sleeve, any change in the flux produced by the current flowing in the coil winding will be opposed by the counter emf of the slug. The flux change will lag behind the primary coil current change by an amount mostly proportional to the volume of copper in the slug.

With the slug at the "front", or "armature-end" of the coil spool, Fig. 1, the flux first leaks inside the slug when the coil winding is energized. Later the transient effect of the slug allows the flux to build out until it acts upon the armature. This time interval, during which the slug is effective in preventing magnetic attraction of the armature, is referred to as "operate time delay". It varies from approximately 100 to 150 milli-seconds maximum for a large size telephone type relay, to not more than 25 to 40 milli-sec max for the miniature relays.

After the magnetic flux is established in full linkage with the slug, the counter emf of the slug also delays collapse of the magnetic field when the coil winding is de-energized. The front-end slug, therefore, is effective in providing both an operate time delay and a release time delay. This characteristic should not be ignored by the relay engineer and user even though such coils are usually employed only for the production of an operate time delay.

SLOW RELEASING COILS. A slug on the end of the coil spool farthest

from the armature, Fig. 2, provides only release time delay as shown. The effects of the counter emf of the slug result in early magnetic linkage of the armature, but the steady state linkage of flux with slug soon occurs if the circuit is maintained for a fraction of a second. On de-energization, the slug counter emf again maintains the flux long enough to produce a release time delay of about 300 to 750 milli-seconds for large size lightly loaded relays, and 40 or 50 milli-seconds for similar miniature relays. Heel end slugs of the type shown in Fig. 3 (*A*) are heavy and relatively expensive.

SLOW ACTING COILS. If the copper slug is closer to the coil core than the winding, its effect is increased. Hence, a smaller volume of copper, in a sleeve closely embracing the core, is required to produce results in time delays that are equivalent to those produced by larger volumes of copper less strategically placed. The sleeve-coil of Fig. 3 (*B*) is one approach to this problem.

When used full length of the coil core, the sleeve coil has some of the effect of both the armature-end slug (slow to operate) coil and the heel-end slug (slow to release) coil. Such relays are generally called "slow acting", usually they are used strictly for slow-to-release purposes, for which they are practically as effective in the production of release time delays as is the heel-end slug coil.

TIMING OF SLUGGED COILS IN GENERAL. In Table I are given the timing limits for different types of telephone type relays that are obtainable under favorable conditions.

The effect of a solid slug, obviously, can be approximated by replacing the slug with a winding in the same location on the coil core, short circuited externally across its terminals either by a solid connection to make its effect permanent as with a solid slug, or through a set of contacts as shown in Fig. 4 wherein the slugging effect of the short circuit is only temporary.

The short circuited front-end section of the relay coil winding, Fig. 4, produces an operate time delay for the usual reasons. But on de-energization, since the relay contacts have opened and removed its slugging effect on release time, a fast releasing, slow-to-operate relay is obtained. For the same winding space the copper volume (density) of wound slugs is less than that of solid slugs, therefore, with wound slugs it is not possible to obtain the maximum time that can be achieved with solid slugs. For example, with all other conditions being the same, the maximum operate time obtainable with a wound slug is only about 45 percent of that obtainable with a solid armature-end-slug occupying the same space.

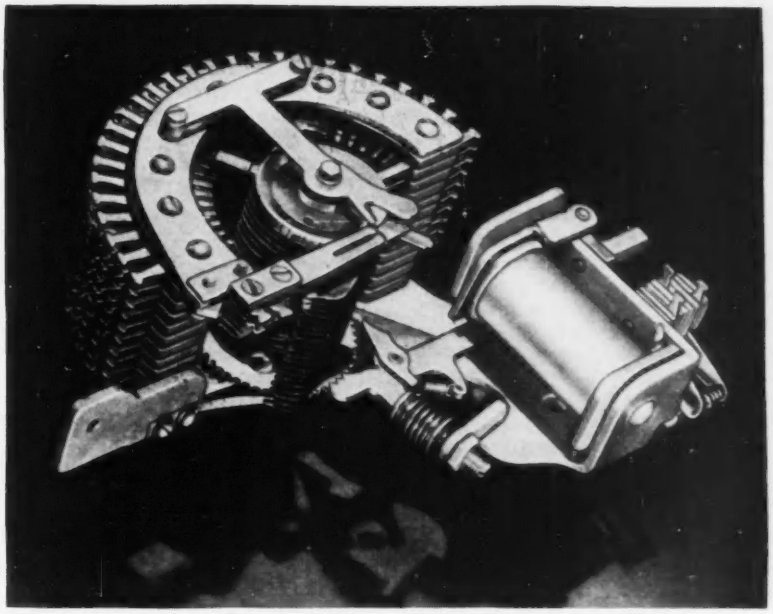


Fig. 6—An indirect drive telephone type single motion stepping switch that returns to initial or home position by continuing on from bank contact point to which stepped.

SHORT LEVER ARMATURE. Prior to the development of the short lever ratio armature, the slow-to-release relay required adjustment of the residual screw to minute settings. Frequently these settings provided no gap beyond that of the plating on the metal of the armature and coil core at the point of contact. Consequently, the wearing away of the plating, the pounding down of the residual setting, or the embedding of the end of the residual screw into the coil core face, soon produced a zero "residual gap". The result was either a relay "sticking up" from residual magnetism, or a release time too long for circuit conditions to tolerate.

These faults were overcome by improving the leverage of the armature. With the short lever armature, as large or larger residual gaps can be set and maintained on slow-to-release relays as on fast acting relays, with the same stability of adjustment, and without sacrifice in timing.

Short lever armatures are suitable for relays that are to be slow to release, such as d-c operated relays with release requirements too long to be met with a "standard ratio" armature and a stable residual gap, a-c operated relays in general, and relays powered from rectified half-wave power sources.

Short lever ratio armatures are to be avoided where relatively fast pulses must be followed, and in applications where such relays are not required. But, conversely, they should be used where they obviously provide an advantage in stability of adjustment, freedom from chatter, or the ability to handle and hold firmly operated a heavier load than could otherwise be accommodated.

STEPPING SWITCHES OR STEPPING RELAYS. To evaluate effectively factors such as speed performance and life, the characteristics of stepping switches or stepping relays should be well understood.

The Minor Switch, Fig. 5, consists in part of a ratchet mechanism operated by a driving electro-magnet, called a rotary magnet, together with a detent arrangement controlled by another electro-magnet called the release magnet. For each electrical pulse closed directly to the rotary magnet, the wiper assembly is moved one step forward on the contact bank. The detent holds the wiper assembly in the position to which it was last advanced until it is either stepped forward again by another pulse or caused to release to the home position.

A coil spring "homes" the wiper assembly. The release action occurs when the detent is removed from the ratchet teeth through energization of the release magnet.

This switch is a "direct drive" stepping switch. It has the disadvantage

of being driven by a force that varies with the electrical power input to the rotary magnet, which in turn varies with the voltage, length of pulse, pulse speed, and "percent make" of the pulse. Thus, the blow struck by the pawl against the ratchet teeth is non-uniform, and the life of the switch is a function of the power input factors. The Minor Switch does not provide the longest life obtainable in stepping switches; under ideal conditions its assured life is only approximately 3½ million, 10-step cycles.

The Rotary Switch, Fig. 6, has only one electro-magnet and the wiper assembly is driven indirectly. Hence, for the most part, the variables of pulse length, percent make, and degree of saturation, are not present. Insofar as the force exerted on the ratchet wheel teeth is concerned, a more controlled and uniform action is obtainable from the Rotary Switch than from the Minor Switch.

When the Rotary Switch coil is energized, the armature is pulled up against the force of a driving spring and is held "cocked" until the pulse is cut off. The wipers remain in their last position during this portion of the pulse. When the armature is released magnetically by de-energization of the rotary magnet coil, the wipers advance under the force of the driving spring and pawl combination. Thus, stepping forces and angles of contact between pawl and ratchet can be held to a uniformly controlled relationship not attainable in the Minor Switch. Therefore, better life and overall performance result.

Lubrication of stepping switches at the factory may differ for different applications depending upon temperature, humidity, dust, and other conditions, therefore, the manufacturer should be advised what is anticipated in regard to frequency of operation, atmospheric conditions, temperature ambients, and life required.

The bank contacting is not ordinarily done through precious or noble metals, as a result certain precautions may be necessary such as shielding from oil, dust, and grit.

In high speed operation, besides the caution that must be exercised in choosing relays with sufficient life and stability of adjustment, the engineer must consider spring loads, fundamental inductance, and other allied factors.

When relays are mounted on machines, the effects of shock and vibration must not be overlooked. The wear produced by vibration of the machine on which the relay is mounted may cause more wear than results from the operation of the relay itself.

Stress-Corrosion Resistance of Wrought Magnesium Alloys

MAGNESIUM ALLOYS, because of their lightness, high strength-to-weight ratio, easy machinability and good weldability, are being used to an increasing extent in the fabrication of electronic components, aircraft parts, portable equipment, auto trailers and numerous other products. While commercial magnesium alloys are usually resistant to ordinary kinds of atmospheric corrosion, they may corrode seriously in certain industrial atmospheres or along the sea coast, especially if they are in direct contact with salt water. H. L. Logan and H. Hessing of the National Bureau of Standards, at the request of the Department of the Navy, have conducted an extensive investigation of the resistance of wrought magnesium alloys to corrosion under stress both in a marine atmosphere and an inland atmosphere.

The Bureau's investigation included both sheet and extruded magnesium alloys. The sheet materials were the ASTM M1, AZ31X, AZ51X, AZ61X alloys and a clad alloy in which a core of AZ31X sheet was sandwiched between two thin layers of the M1 alloy; extrusions studied were the ZK-60 (Dow), AZ61X, AZ80X, and AZ-80X-HTA materials. Tests were made with ½ in. reduced-section standard ASTM tension specimens for sheet metals, except that the grip ends were one or 1¼ in. wide, when possible, instead of the standard ¾ in. The purpose of this added width was to minimize failures caused by stress corrosion around bolt holes.

The specimens were exposed while stressed in tension at the National Bureau of Standards in Washington and in a marine atmosphere at Hampton Roads, Virginia. Unstressed specimens were subjected to the same corrosive conditions so that the effect of stress in increasing corrosion damage could be evaluated. Laboratory tests were also made by continuous immersion of stressed specimens in a sodium chloride-potassium chromate solution and by intermittent immersion in a 0.01 percent sodium chloride solution. Specimens stressed by weighted levers were supported vertically in the test solutions in sealed glass cells. For the intermittent immersion tests, the corroding solution was raised into the cells with compressed air, wetting the specimens four times per hour.

"Threshold" stresses, defined as the maximum stresses that materials can withstand without failure when continuously immersed in the corroding medium for a fixed period of time, were determined as a measure of corrosion resistance in the sodium-chloride-potassium chromate solution. Resistances to stress corrosion in intermittent immersion and weather exposure tests were evaluated from exposure periods at given stresses.

Susceptibility of magnesium alloys to stress corrosion in the weather increased with aluminum content up to about 6.5 percent. It was found that specimens stressed at the Washington weather exposure site failed in shorter periods of time than those exposed to the same stress in a marine atmosphere. Damage to unstressed specimens at the Washington site was small as measured by losses in tensile properties. Thus, the AZ31-X-h material, after being exposed unstressed for 750 days, had a tensile strength of more than 94 percent of that of the unexposed materials, and its elongation was still more than 85 percent of the original value.

Of all the materials studied, the Ml-clad AZ31X-h alloy was the most resistant to stress corrosion. Specimens of this alloy were exposed in a marine atmosphere while stressed to 30,000 psi (90 percent of yield strength) for 500 days without failure.

Among the bare alloys, the Ml-h sheet and the ZK-60 extruded material were outstanding in stress-corrosion resistance. After 1,175 days of exposure at Washington, under a stress of 16,000 psi (55 percent of yield strength), the Ml-h sheet alloy had not failed. First failures of the ZK-60 extruded material exposed in the same locality under a stress of 20,000 psi occurred only after 580 days, while specimens of this alloy stressed to 18,000 psi had not failed after more than 1,010 days.

Of the remaining alloys, the bare AZ31X-h material proved the most resistant to stress corrosion. However, specimens of this material stressed to 16,000 psi (about 50 percent of yield strength) and exposed at Washington failed after an average exposure period of only 151 days. The AZ51X, AZ61X and AZ80X alloys were all susceptible to stress corrosion when exposed under stresses of 20,000 psi or more. The extruded AZ80X was more resistant to stress corrosion in the heat-treated and aged condition.

ENGINEERING ABSTRACTS

FOREIGN AND DOMESTIC ARTICLES

New Lightweight Diesel Engine

Foreign abstract from Mechanical World (London) p 313, March 24, 1950.

A NEW HIGH-PERFORMANCE DIESEL engine possesses a number of outstanding features. Especially good is its high power-to-weight ratio; with a bore and stroke of 5.4 in x 6 in., it has a displacement of 1,099 cu in., and develops 320 bhp at 2,400 rpm; its dry weight with all essential accessories and ready for installation, is only 1,500 lb. This has just been announced by The Rover Company Limited, of Birmingham.

With a 4-stroke cycle and overhead valves, the engine has eight cylinders arranged in two banks inclined at 60 deg. Compression ratio is 16.5 to 1. At the front end are the auxiliaries: centrifugal-type water pump, twin fuel feed pumps and gear-type oil pressure and scavenge pumps. A fan drive shaft is coaxial with the crankshaft and belt pulleys can be attached. All accessories are driven through a spring drive for smooth operation. At the rear is a gear case, which drives the injection pump, and a power take-off shaft for driving accessories.

Aluminum alloy (D.T.D.133) is used for the crankcase which is adequately ribbed for rigidity. The case carries five main bearings of the easily replaceable steel-backed lead-tin plated type. The rear main bearing is extra long to support a clutch and incorporates thrust faces for axial location of the crankshaft. Each bearing cap is held in position by two vertical studs. The crankcase is further stiffened by the addition of the aluminum alloy cylinder blocks and heads, held to the crankcase by 20 (10 per block) long studs which take all tensile loading resulting from combustion pressures. Lateral bolts passing through the side walls and main bearing housings relieve the crankcase of dynamic stresses resulting from the inclined banks of cylinders.

The front end of the crankshaft carries a friction type torsional vibration damper of Rover manufacture to provide smooth running throughout the whole speed range. Made from chrome molybdenum steel, the nitrided four-throw, 180 deg spaced crankshaft incorporates integral balance weights. All crankpins and journals are bored for lightness. The connecting rods are of the plain and forked type. They are machined all over and finished by shot peening to provide maximum resistance to fatigue. The wrist-pin bearings consist of floating bushings and the pin is also fully floating in the piston. This assembly is lubricated by an oil splash. The crankshaft bearings are of lead bronze, plated with a tin-lead flash and are carried in thin steel shells.

The pistons are carefully proportioned and have a specially designed crown to insure efficient combustion. They are anodised and impregnated with the oil to ensure a low rate of wear, and carry three compression and two scraper rings above the wrist pin, and one scraper ring below. The pins are held by two retaining rings.

Much thinner than usual, the cylinder liners are of forged manganese steel and are chromium plated at the top of the bore so that wear is reduced to an absolute minimum. They are readily detached from the cylinder block, being of the wet type, sealed by heat and oil resisting rings.

Three valves per cylinder are employed, two intakes of $1\frac{3}{8}$ in. diameter and one exhaust of 2 in. diameter. The exhaust valves are sodium cooled; they are Stellite tipped and Brightray seated. Sil-chrome is used for the inlet valve seats, while an austenitic steel is used for the exhaust valve and seat. All valve seats are screwed and shrunk into the heads. To facilitate easy fitting and removal of the valve guides, they are tapered on the outside diameter and fit into a tapered hole in the cylinder head. Cast iron and phosphor bronze are used for the inlet and exhaust valve guides respectively. Corrosion-protected twin valve springs are fitted to each valve.

The spherical pre-combustion chamber is in two parts, the top half carrying the injector while the lower serves as a hot plug; they are held together, and to the cylinder head, by three studs, and sealed by copper washers.

Overhead camshafts are used, the cams operating the valves through rocker arms. The camshafts are driven from the front end of the engine through bevel gears and provision has been made for expansion of the cylinder blocks and for the minute movements which occur in all reciprocating engines between the cylinder heads and the crankcase. The uneven torque, always to be found in a camshaft drive, is absorbed by the use of a torsion spring shaft provided with a friction damper.

The lubrication system consists of a separate sump tank, a coarse filter, two fine cloth-type full flow filters, and a pressure regulating unit. This unit divides the supply, supplying at high pressure to the crankshaft journals and end bearings and at low pressure to the camshafts, rocker bearings and some of the auxiliary gearing. Both systems drain into the sump which is kept "dry" by a scavenge pump.

The centrifugal-type coolant pump with a forged brass impellor is keyed to a stainless steel spindle and is driven by bevel gears. All water passages in aluminum alloy components are coated with Bakelite varnish to ensure freedom from corrosion.

The fuel is injected by C.A.V. equipment of the constant stroke type employing one pumping element for each cylinder. It is fitted with a pneumatic governor. The pintle valve injection nozzles are simple to maintain and are not readily susceptible to fuel oil impurities.

Titanium Alloys For High Temperature

Foreign abstract from "High Temperature Alloys Based on Titanium Carbide", by E. M. Trent, A. Carter and J. Batemen, page 111 of August 1950 issue of Metallurgia (Manchester, England).

IN THE SEARCH FOR MATERIALS capable of withstanding severe high temperature operating conditions, many different types of materials have been considered. The "hot strength" of cemented carbides is utilized in metal cutting operations and investigations of the properties of alloys

based on titanium carbide and containing chromium and nickel or cobalt, have shown them to possess many of the properties desirable in gas turbine materials. The low specific gravity of alloys containing a large proportion of titanium carbide make them particularly suitable for turbine rotor blades. Hardness tests on cemented carbides at high temperatures show a relatively small drop in hardness at temperatures up to 750 C. Titanium carbide has the added advantage of a low specific gravity.

Alloys of a range of compositions, containing titanium carbide, chromium carbide and iron, nickel or cobalt, have been patented. This paper gives the results of some of the tests carried out on alloys based on titanium carbide and containing, also, chromium and nickel or cobalt.

Using the Slide Rule

From "Slide Rule Tricks" by Henry Eckhardt, Chemical Engineering, July 1950, page 119.

A METHOD IS OUTLINED for obtaining the solution of several physical and engineering functions by the slide rule. Following relations are solved in the common engineering units.

Wire Gage—Wire Diameter
Reynolds Number function—Reynolds Number
Hydraulic friction factors—Reynolds Number

Solutions are expressed in terms of the slide rule settings.

EXAMPLE: The relation between the dia. and area of a circle is given by

$$a = d^2/4$$

The equivalent slide rule setting is

$$C(1)/D(d) = A(a)/B\ (/4)$$

The capital letters denote the scales of the slide rule; the figures in parentheses denote the corresponding scale reading.

Wire Gage-Wire Diameter

The wire diameters in the AWG or B & S gage for successive sizes are in the ratio $92^{1}/_{39}$. The full scale ranges from No. 0000, the largest, to No. 40, the smallest. The method which gives the diameter for all sizes in a single setting is based on a hypothetical gage of 49.88 . . . . which would be 1 mil in diameter. The gage point 1159 on the D scale is used. Accurracy is limited only by slide rule precision.

Let d = wire diameter in mils, G = gage number (algebraically). (Thus, a No. 0000 wire is -3 on an algebraic scale.) Then the slide rule setting is

$$C(1)/D(1159) = C(49.88-G)/LL3(d) \quad (1)$$

The left index of the C scale is used for all settings.

EXAMPLE: Find the diameter of a No. 10 B&S gage wire.

Set the left index of scale C opposite the point 1159 on scale D. 49.88 minus the gage size, 10, is 39.88. Then opposite 39.88 on scale C, find 102 on scale LL3. The wire diameter for a No. 10 B&S gage wire is thus 102 mils. (See Fig. 1.)

EXAMPLE: Find nearest B&S gage size for a 0.020-in. diameter wire.

Same setting of slide as for above example. Opposite 20 on scale LL3, find 25.84 on scale C. Then the gage size, G, equals 49.88 minus 25.84, or 24.04. Thus the given size is slightly smaller than a No. 24 gage, the latter being 20.1 mils diameter. (See Fig. 2).

Reynolds Number Function

The Reynolds Number function is used to determine the fluid friction head by means of the modified Fanning equation.

$$F = \phi N_{Re}\, Lu^2/D \quad (2)$$

where ϕN_{Re} = the Reynolds Number function, F = fluid friction head, feet of fluid flowing, L = length of pipe, feet, D = inside diameter of pipe, feet, u = velocity of stream, feet per second.

Reynolds Number function is a condensation of the fluid friction factor, f, and the constant term $2/g_c$, which occurs in the original form of the Fanning equation.

$$\phi N_{Re} = 2f/g_c \quad (3)$$

where $g_c = 32.17$ lb.$_m$ ft./lb.$_f$ sec.^{2} is the dimensional constant. Following this section a similar condensation of factors is suggested which will expedite the calculations for determining the Reynolds Number itself.

Pipe friction factors vary with the state of flow—viscous or turbulent—and the nature of the interior pipe surface. Viscous, or streamline flow, is considered to hold up to about N_{Re} 2,000, and stable turbulent flow from about N_{Re} 4,000 upwards.

In the viscous flow region the fluid friction is independent of the internal pipe surface conditions. The equation is

$$\phi N_{Re} = 1/N_{Re} \quad (4)$$

In the region of turbulent flow the fluid friction is dependent on surface conditions inside the pipe. The modified slide rule setting for the turbulent range of N_{Re} from 4,000 to 2,200,000 is

$$C(1)/LL3(N_{Re}/100) = C(10^4\, \phi N_{Re})/LL3(GP) \quad (5)$$

where GP denotes the gage point employed. For all cases, the right hand index of scale C is employed. The factor 10^4 set before the Reynolds Number function on scale C locates the decimal point properly. The C scale is assumed to range from 1 to 10. However, in actual use it is simplest to disregard this factor, since all values of the Reynolds Number function in the turbulent range have three zeros after the decimal point.

As a first approximation, a gage point of 11 is generally applicable for practical calculations. Useful range N_{Re} 4,000 to 1,000,000.

EXAMPLE: Find the Reynolds Number function for a Reynolds Number of 50,000.

Set the right C index opposite 500 on the LL3 scale. Opposite the gage point 11 on the LL3 scale, find 386 on the C scale. Place the decimal point followed by three zeros before this reading and obtain 0.000386 for the Reynolds Number function. The computed value is 0.000382; consequently, the value found by slide rule is only about 1 percent greater. (See Fig. 3).

Friction Factors "f"

Since the use of the Reynolds Number function is more convenient than the use of the friction factors, the latter are necessary only for correlating calculations with other work; consequently, for such occasional use the most convenient way is to determine the Reynolds Number function in the usual way with the slide rule using either Eqs (4) or (5) depending on the conditions of flow, and multiply this value by the conversion factor 16.09, since by Eq (3).

$$f = \tfrac{1}{2}\, g_c\, \phi N_{Re} = \tfrac{1}{2}\, 32.17\, \phi N_{Re} = 16.09\, \phi N_{Re} \quad (6)$$

The f referred to here is the Fanning friction factor used in chemical engineering treatises. The f used in hydraulic treatises is applicable to the Darcy equation and it is exactly four times as large as the former f. Thus

$$f_{Darcy} = 4f_{Fanning} \quad (7)$$

For most practical purposes, the conversion factor for converting from Reynolds Number function to friction

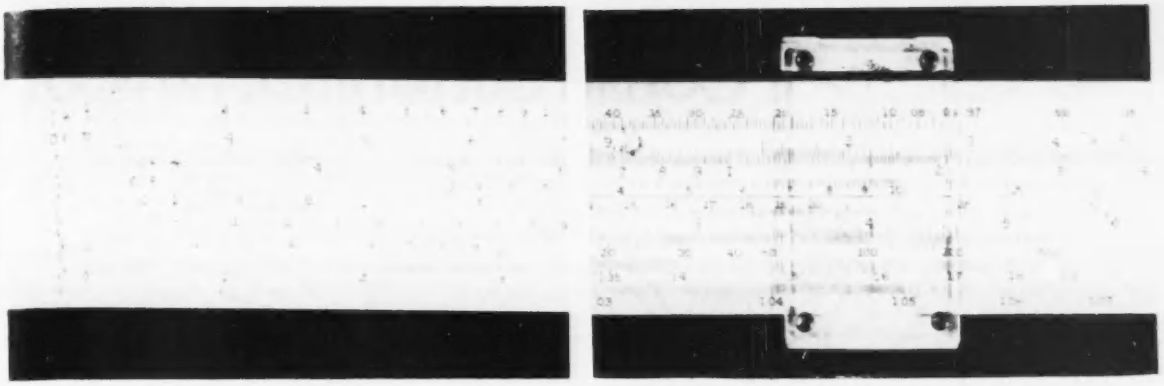


Fig. 1—No. 10 B&S wire is 102 mils in diameter

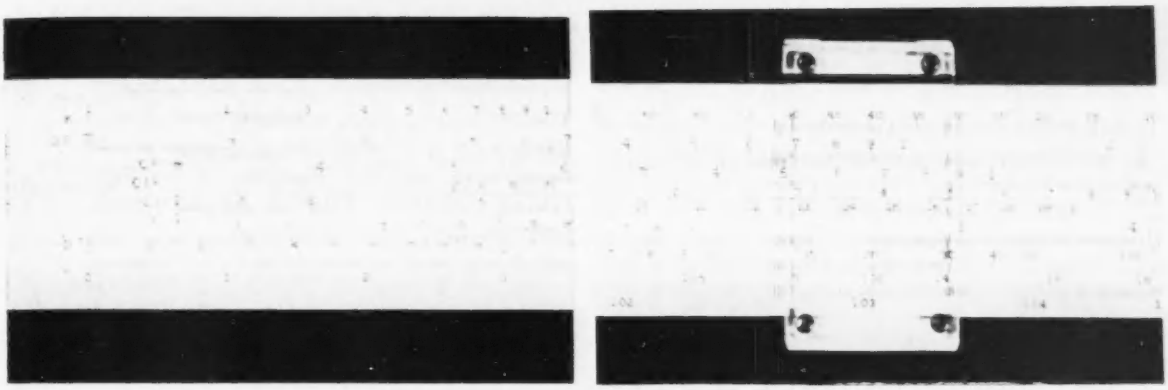


Fig. 2—0.020 in. wire is No. 24 B&S gage.

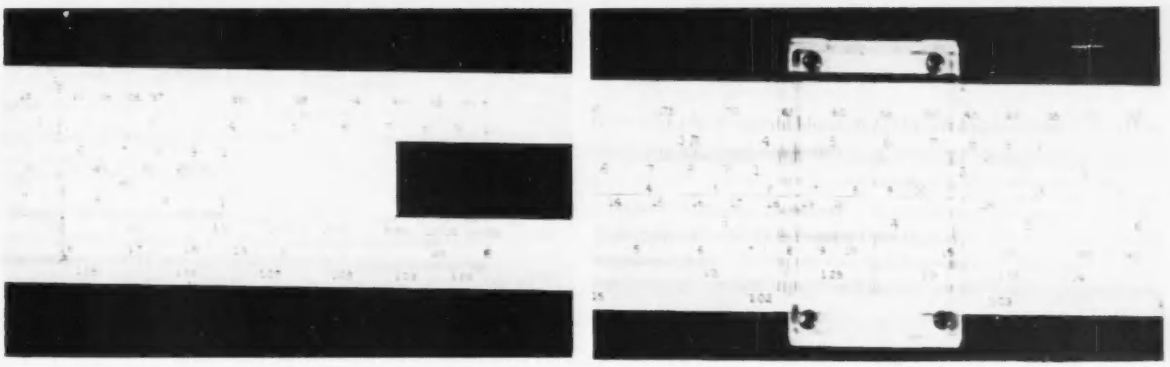


Fig. 3—Reynolds Number function is 0.000386 for a Reynolds Number of 50,000.

factors may be taken as 16 and 64 for use with the Fanning and Darcy equations, respectively.

Computing the Reynolds Number

A hint is given of a method for computing the Reynolds Number quickly. The work of computation is shortened by combining all the conversion factors into a single constant suitable for the system of units employed. Since the factor is constant the Reynolds Number can be found directly from the slide rule. The crux of the method is to take water as a reference liquid and assign unit specific gravity and unit viscosity to it. Water at 68 F. deg. has a viscosity of 1 centipoise; consequently, the specific viscosity of any fluid (compared to water at 68 deg. F.) is numerically equal to its viscosity expressed in centipoise, the latter unit being the most commonly employed. Similarly, specific gravity is numerically equal to the density expressed in the metric system.

Consider the flow of water at 68 deg. F. at moderate rates where Q = flow gallons per minute and d denotes the inside pipe dia., in in. Then,

$$N_{Re} = 3,160\,Q/d \qquad (8)$$

For any liquid at any temperature multiply Eq (8) by the specific gravity and divide by the specific viscosity of the liquid, thus

$$N_{Re} = 3,160\,Q(SG)/d(SV) \qquad (9)$$

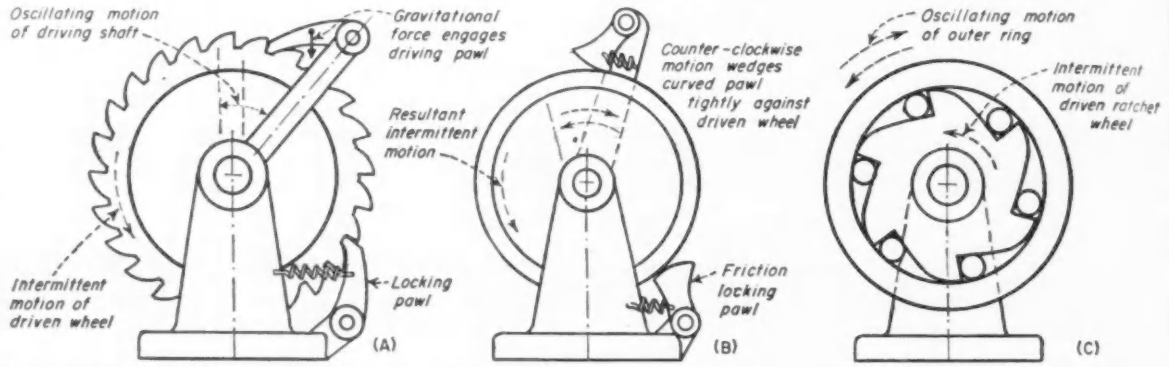
where SG = specific gravity and
SV = specific viscosity.

EXAMPLE: What is the Reynolds Number for a 40 percent sucrose solution at 20 deg. C. flowing at the rate of 80 gpm. in a 3-in. Sched. 40 pipe?

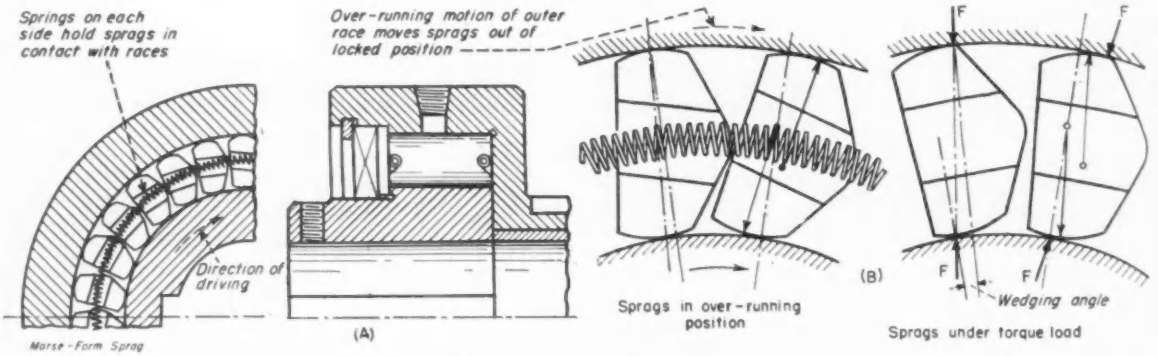
Original data: I.D. of 3-in. Sched 40 pipe is 3.068 in.; SV of sucrose solution is 6.223; SG of sucrose solution is 1.1764. Substituting these in Eq (9).

$$N_{Re} = (3,160)(80)(1.176)/(3.07)(6.22) = 15,570 \qquad (10)$$

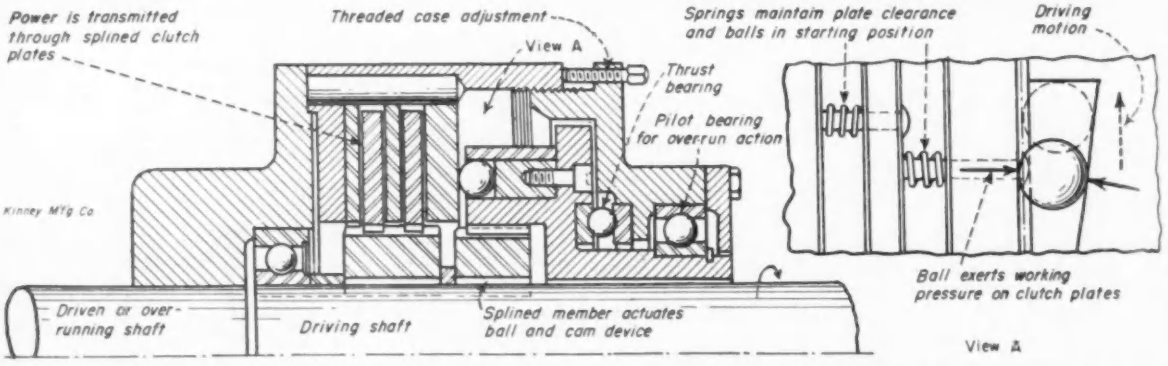
Construction Details of



1 Elementary over-riding clutches: (A) Ratchet and Pawl mechanism is used to convert reciprocating or oscillating movement to intermittent rotary motion. This motion is positive but limited to a multiple of the tooth pitch. (B) Friction-type is quieter but requires a spring device to keep eccentric pawl in constant engagement. (C) Balls or rollers replace the pawls in this device. Motion of the outer race wedges rollers against the inclined surfaces of the ratchet wheel.



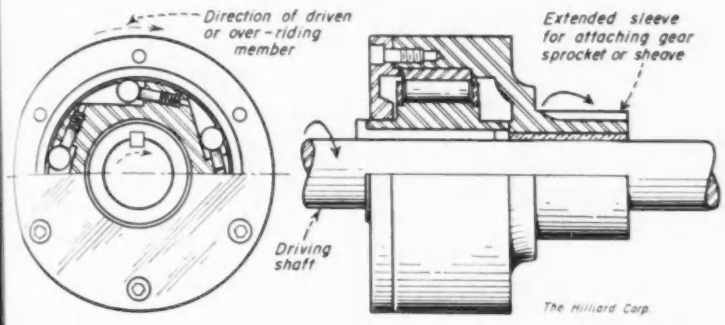
4 With cylindrical inner and outer races, sprags are used to transmit torque. Energizing springs serves as a cage to hold the sprags. (A) Compared to rollers, shape of sprag permits a greater number within a limited space; thus higher torque loads are possible. Not requiring special cam surfaces, this type can be installed inside gear or wheel hubs. (B) Rolling action wedges sprags tightly between driving and driven members. Relatively large wedging angle insures positive engagement.


Power is transmitted through splined clutch plates
Threaded case adjustment
View A
Springs maintain plate clearance and balls in starting position
Driving motion
Thrust bearing
Pilot bearing for overrun action
Kinney MFg Co
Ball exerts working pressure on clutch plates
Driven or over-running shaft
Driving shaft
Splined member actuates ball and cam device
View A

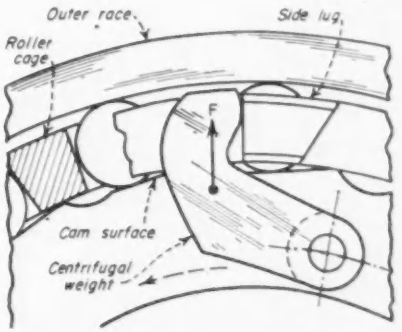
6 Multi-disk clutch is driven by means of several sintered-bronze friction surfaces. Pressure is exerted by a cam actuating device which forces a series of balls against a disk plate. Since a small part of the transmitted torque is carried by the actuating member, capacity is not limited by the localized deformation of the contacting balls. Slip of the friction surfaces determine the capacity and prevent rapid, shock loads. Slight pressure of disk springs insure uniform engagement.

Over-Riding Clutches

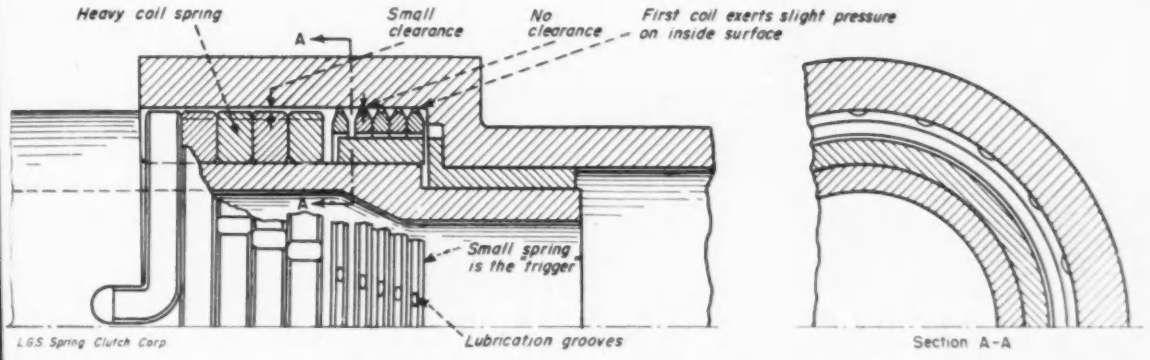
A. DeFEO
Design Engineer
Wright Aeronautical Corporation


Direction of driven or over-riding member
Extended sleeve for attaching gear sprocket or sheave
Driving shaft
The Hilliard Corp.

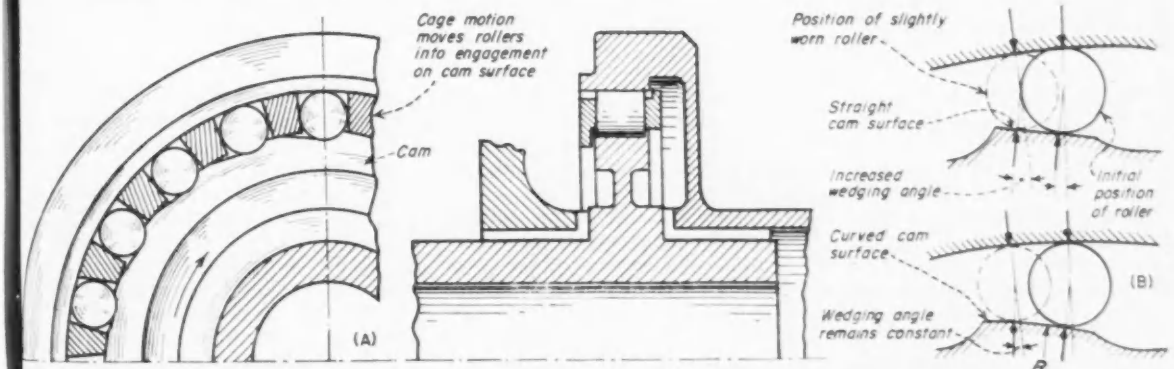
2 Commercial over-riding clutch has springs which hold rollers in continuous contact between cam surfaces and outer race; thus there is no backlash or lost motion. This simple design is positive and quiet. For operation in the opposite direction, the roller mechanism can easily be reversed in the housing.



3 Centrifugal force can be used to hold rollers in contact with cam and outer race. Force is exerted on lugs of the cage which controls the position of the rollers.



4 Engaging device consists of a helical spring which is made up of two sections: a light trigger spring and a heavy coil spring. It is attached to and driven by the inner shaft. Relative motion of outer member rubbing on trigger causes this spring to wind-up. This action expands the spring diameter which takes up the small clearance and exerts pressure against the inside surface until the entire spring is tightly engaged. Helix angle of spring can be changed to reverse the over-riding direction.

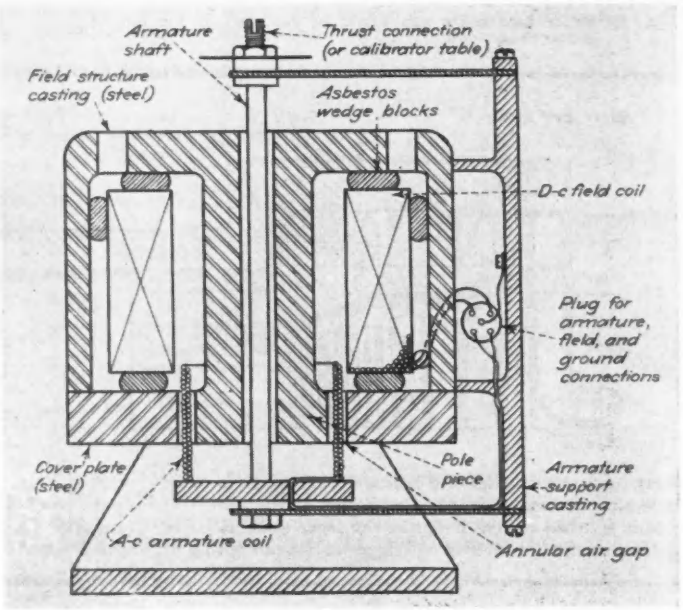


7 Free-wheeling clutch widely used in power transmission has a series of straight-sided cam surfaces. An engaging angle of about 3 deg is used; smaller angles tend to become locked and are difficult to disengage while larger ones are not as effective. (A) Inertia of floating cage wedges rollers between cam and outer race. (B) Continual operation causes wear of surfaces; 0.001 in. wear alters angle to 8.5 deg. on straight-sided cams. Curved cam surfaces maintain constant angle.

Applying Shakers for Product Vibration Analysis

Fig. 1—Essential elements of the electrodynamic shaker. D-c current in field coil causes a magnetic flux of high density to flow across the annular air gap and through the armature coil. A-c current in armature coil reacts with the magnetic flux to produce an alternating force on the armature shaft and sets it in motion.

Basic characteristics of electrodynamic shakers and their role in testing mechanical components.



ROBERT C. LEWIS
The Calidyne Company

THE ELECTRODYNAMIC SHAKER is a testing device for producing an alternating force that vibrates or shakes objects attached to its armature. The characteristics of this device are such that the magnitude, waveform and frequency of the force produced can all be controlled over wide ranges, and permit direct measurement of mechanical force and power delivery. In these respects, the shaker differs substantially from mechanical devices used to rattle, shake or jar parts and assemblies.

Taking into account all of the many fields in which electrodynamic shakers are and can be used, there are four general classifications of test work that define the more common uses:

(1) Determination of the Dynamic Response Characteristics of Mechanical Structures—This first category involves the determination of natural mode frequencies and the excitation of the natural mode responses to allow definition of the mode shapes and damping characteristics. Tests of this sort are diagnostic in nature and are intended to furnish specific performance data for the calculation of behavior in complicated dynamic situations; to verify theoretical calculations of dynamic response; or, in simpler cases to locate specifically the cause of objectionable noise or breakage.

(2) Fatigue Testing Structures, Mechanisms or Components at Mechanical Resonance—Tests in this class are of the endurance type which, in suitable cases, permit the application of high stresses at high frequency by using the resonant frequency of the part under test.

(3) Subjecting Components or Assemblies to Vibratory Accelerations—Here a relatively compact instrument or mechanism is attached to the armature and shaken by "brute force." The

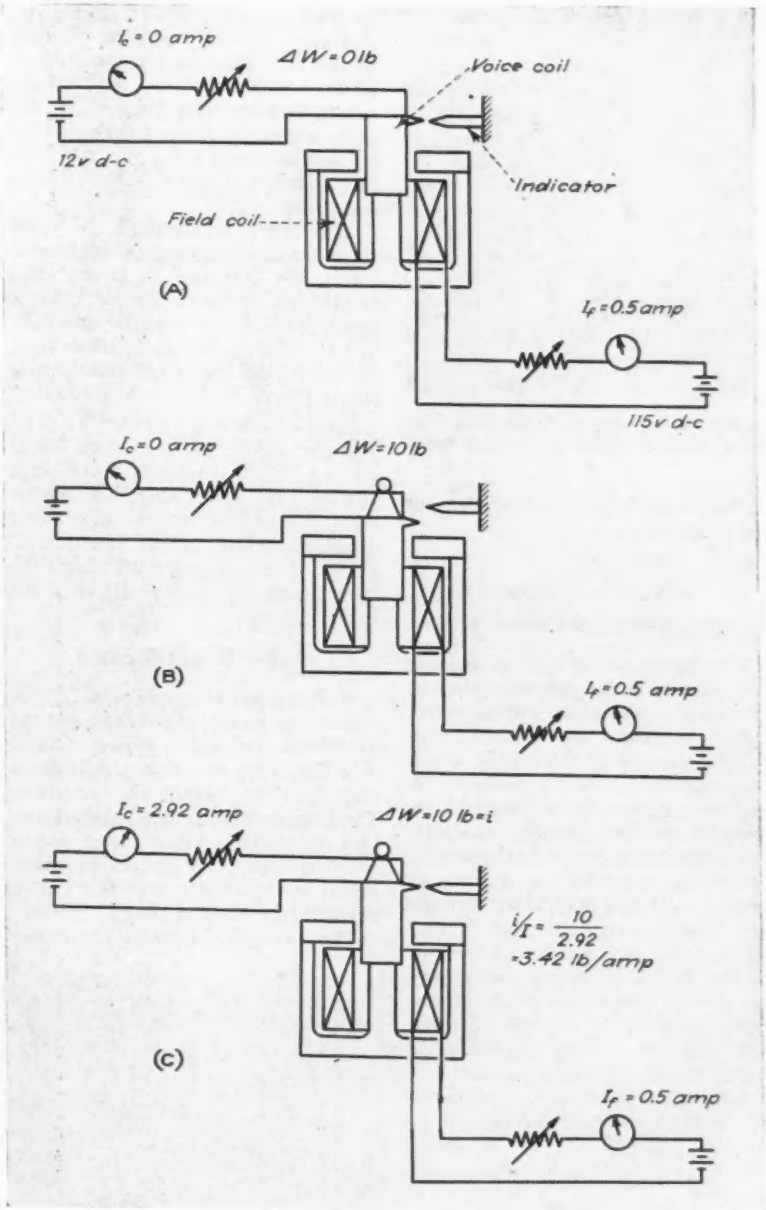


Fig. 2—Determination of the basic shaker force-current ratio by dead-weight calibration.

frequency range is chosen such that resonances of small elements or components are excited to determine the ability of the complete instrument or mechanism to withstand vibrations that are defined in terms of the acceleration level and frequency range. The acceleration that a shaker can produce is equal to the shaker force divided by the weight shaken and expressed in g units.

(4) Subjecting Elastic Elements to Repeated Flexure—Use of the electrodynamic shaker to deflect elastic elements by "brute force" is possible, but more frequently the versatility of the shaker is utilized by adding sufficient mass to the elastic system to put it in resonance at a convenient test frequency, as in the second classification above. Under this condition, the deflections of the elastic member can be increased by a factor of from 20 to 150 times that obtained by the direct action of the shaker force.

Shaker Components

The essential elements of the electrodynamic shaker illustrated in Fig. 1 are the field structure and the armature. The field structure is a pot-shaped piece with a center post, capped by a pole piece and cover plate to complete the magnetic circuit. A uniform circular air gap is left between the cover plate and central pole piece and a high-density magnetic flux established across it by a d-c field coil. The armature consists of a coil suspended centrally in the air gap, and the structural members necessary to transmit the forces that are generated by the armature coil to objects secured to the armature shaft.

Forces are produced between the field structure and the armature when a current flows through the armature coil suspended in the magnetic field. The force on the armature is the sum of the forces acting on each coil conductor in the magnetic field. When converted to pound-inch-second units, the basic force relationship is:

$$i = 0.885 \times 10^{-7} BLNI \qquad (1)$$

where i = force, lb
B = flux density, lines/in.^{2}
L = length of single armature turn, in.
N = number of turns in flux gap
I = conductor current, amp

For any particular shaker design, L and N are fixed properties of the armature and Eq (1) indicates that the force produced by such an armature is proportional to the current flowing through its conductors *if the field strength is constant*. Assuming for a moment that this is so, it can be seen that the shaker force is directly related to the current. The force-current ratio is therefore a basic and useful property of the electrodynamic shaker. When derived from Eq (1), this relationship becomes:

$$i/I = K, \text{ lb/amp} \qquad (2)$$

This ratio can be found experimentally by measuring the current required to lift a known weight, as indicated by Fig. 2. All current measurements are made with the armature in the same position relative to the field structure, so that the restoring forces of the armature support system remain constant. Once this force-current ratio is known, it is a simple matter to measure the mechanical force produced with a meter of suitable scale range.

This calibration procedure utilizes the relationship in Eq (1) between direct currents and static forces. When the magnitude of the current varies with time—as in the case of an alternating current—the force will vary in exactly the same way. The convenience of measuring rapidly the varying forces by monitoring the electrical current to the shaker is therefore ap-

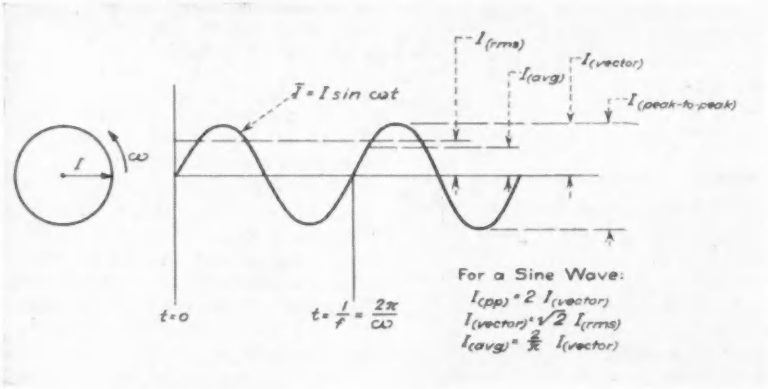


Fig. 3—Designation of various measures of an alternating current, *I*, the time trace of which is illustrated as the projection of a rotating vector or a sine wave.

parent. Another convenience derived from this relationship is the relatively pure sinusoidal force excitations that are obtainable with standard equipment used in the fields of electrical communications and power engineering.

Measurement of Alternating Quantities

Non-standard practice in designating or defining the magnitudes of alternating forces, motions, currents or other quantities is sometimes the source of confusion for vibration engineers. Fig. 3 illustrates graphically a sinusoidally oscillating (or alternating) current, the instantaneous value of which is *I*. The magnitude of this current at any instant is $I \sin \omega t$. This quantity can be thought of as the projection of a vector of length *I* rotating at an angular velocity of ω, as indicated at the left in Fig. 3. For the sake of clarity, this maximum deviation of the current from a zero or mean value is best referred to as the *vector* current, or $I_{(vector)}$ thereby avoiding confusion with average, rms and peak-to-peak values of current.

According to Eq (1), if a sinusoidally varying current is produced in the shaker armature, a force proportional to the current is developed. Thus, the force wave is represented by the current wave as shown in Fig. 3, and the alternating force can be measured in terms of $i_{(vector)}$, $i_{(pp)}$, $i_{(rms)}$ or $i_{(avg)}$. Since the maximum value of the force, or $i_{(vector)}$, is generally of primary interest, shakers are usually rated in the terms of this quantity. On the other hand, if the force is measured by use of Eq (2) and an ammeter, special attention must be given to clarify the current designation. The ammeter reads current in terms of $I_{(rms)}$ and therefore the force will be measured in terms of $i_{(rms)}$ or

$$K \times I_{(rms)} = i_{(rms)}, \text{ rms lb} \qquad (3)$$

If the waveform is sinusoidal, or nearly so

$$i_{(vector)} = \sqrt{2}\, i_{(rms)} \qquad (4)$$
$$= \sqrt{2}\ K \times I_{(rms)}, \text{ vector lb}$$

Variation of Force-Current Ratio

The major assumption in arriving at the fundamental proportionality of force and current in the electrodynamic shaker, expressed in Eq (2), was that the strength of the field in which the armature moves is constant. A field current monitor and control are provided so that the total magnetic flux in the circuit can be held constant.

Even with constant total flux, an armature coil that must move through large mechanical displacements may not have the same number of turns in the flux field at all times, thereby giving rise to deviations of the force-current ratio as a function of armature position. This deviation is caused in part by fringing of the flux to fill a space larger than that of the gap between the pole pieces, and in part by distortion of the magnetic field by the current flowing in the armature.

Thus, the requirements for large mechanical displacement, maximum force output, lightweight armature and good waveform call for a balancing of these various factors to insure satisfactory shaker performance.

Rating of Shakers

An electrodynamic shaker is properly rated in terms of the maximum force it can deliver continuously without being overheated by the required current. This rating alone is not sufficient for "brute-force" applications, in which the ratio of rated force to armature weight, or maximum acceleration, should also be stated. Fig. (4) shows the significance of this rating. A shaker of a certain size has a definite force limit within reasonable limits of temperature rise. The magnitude of this force depends upon the efficiency of the field structure, the construction of the armature coil and other related design factors. Larger forces, for the same allowable temperature rise, can be obtained either by the use of a larger shaker or several shakers of the same size.

The latter arrangement is sometimes the more economical because of lower first cost due to standardized production, and also because there is more flexibility in applying the individual units—either simultaneously in different locations or collectively on a single job.

Forces exceeding the rated value of a shaker can be obtained with forced cooling. Short periods of operation at overload are permissible, but continuous overloading results in eventual armature failure. Strict interchangeability of replacement parts makes the latter procedure economically attractive in some cases.

Shaker Power Supplies

A shaker power supply must furnish a source of filtered d-c current for the field excitation and a source of variable frequency current controllable in magnitude to supply the armature. The field supply for small shakers can be a battery or a rectifier-filter system powered from an a-c line. Large shaker fields are generally supplied by a motor driven d-c generator.

The variable-frequency armature

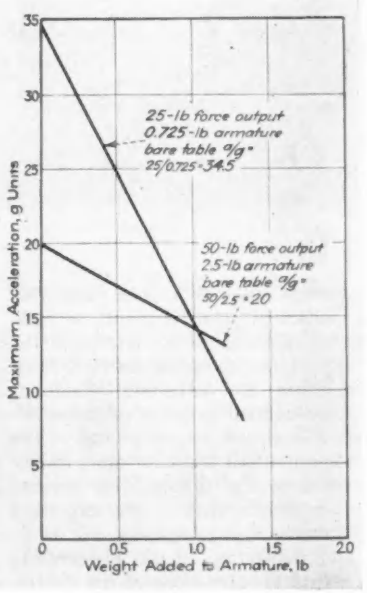


Fig. 4—Bare-table acceleration is a significant rating factor. A 25-lb shaker with a light armature performs better than a 50-lb unit having a heavy armature.

supply can be either one of two general types: (1) Electronic, consisting of a power amplifier and a variable frequency oscillator, and (2) electromechanical, consisting of a suitable alternator driven at speeds controllable over a wide range of frequency output.

The first system is particularly applicable for supplying limited but reasonably large amounts of power to one or two shakers. It is particularly convenient in providing fingertip control of both force amplitude and frequency by means of two dials on the oscillator.

The second system is useful in applications where durability and freedom from maintenance of the motor-alternator system are desirable, and where higher power output is required for either single or multiple-shaker applications. Force amplitude and frequency control are again most convenient, but the rate at which frequency can be changed is sometimes limited by rotational inertia in the rig.

The design of a power system to supply a shaker that can be called upon to drive almost anything at almost any frequency poses many problems. These arise in connection with the requirement of high force availability (current delivery) over a wide frequency range.

Estimating Shaker Performance

The electro-dynamic shaker being a special kind of electric motor, some aspects of its performance limitations can be illustrated by analogy or contrast with motor operation. The two devices are similar in their function of converting electrical power to mechanical power. The conventional motor converts electrical currents to forces that cause undirectional mechanical motion or rotation, whether a-c or d-c powered. By contrast, the shaker is a device that produces forces of changing direction.

A simple statement of the performance obtainable from a shaker is impossible, because performance is dependent on the mechanical load. To ask what performance is obtainable from a 300-lb shaker without stating the properties of the system to be shaken is like asking what power a 5-hp motor will deliver when driving a blower. As a further analogy, the motor can be used to measure the blower torque-rpm characteristic, just as the shaker can be used to measure the force-velocity characteristic of a dynamic load.

In the case of the motor-blower system, assume a variable-speed d-c motor is driving a blower fan. By virtue of the relationships of Eq (1), it is possible to calibrate the motor torque in terms of its current drain. If a current producing a torque of T lb-in. is fed to the motor, its speed will increase until the torque required to drive the blower is just equal to T, the torque produced. If the resultant rpm is measured, one point on the torque-rpm characteristic can be determined.

Similarly, shaker performance depends on the load characteristic. Assume, for comparison, that a reciprocating pump or a damper (which absorbs real power as opposed to reactive power), is driven by the shaker armature. The armature supply is set at f cycles per second and a current (measured in rms amperes) is fed into the circuit to produce $i_{(rms)}$ pounds of force. Under these conditions, the stroke of the pump or damper will increase until the force required to drive the pump back and forth is just equal to the alternating force measured in terms of $i_{(rms)}$.

Although the most tangible measure of the pump stroke is the *displacement* amplitude, a more useful measure in dynamics is the *velocity* amplitude. In either case, there is a variation with time, so that care must be taken to specify whether values are in terms of vector, peak-to-peak, rms or average quantities. Fig. 5 shows the relationship of velocity amplitude to the displacement amplitude.

Applied to the shaker, the velocity of the displacement vector is $2\pi fs$ in magnitude and is measured in units of inches per second. The magnitude in terms of rms velocity is:

$$e_{(rms)} = 0.707 \times 2\pi fs\ (_{vector}),\ \text{rms in./sec} \qquad (5)$$

The same velocity is obtained at a smaller displacement amplitude if the frequency is increased.

In the dynamic case, the power is found from the relationship:

$$P = e_{(rms)}\, i_{(rms)} \cos\theta,\ \text{in.-lb/sec} \qquad (6)$$

where θ is the phase angle between force and velocity (zero when the load is non-reactive).

Although the mechanical load of a resonant structure presents a unity power factor ($\cos\theta = 1$), due to the wide range of frequencies over which the shaker may operate, the electrical inductance of its armature can influence the design of the power supply considerably. In addition, the methods of "brute-force" shaking require the delivery of large forces at very poor power factor. Consequently, the actual power delivered is small in comparison to the shaker losses, which also vary with frequency.

Shakers are generally designed for an armature travel of from 0.1 to 0.5 in. vector displacement and for a stated force delivery. Eqs (5) and (6) show that if the force and displacement amplitude are assumed constant at their maximum values, the power required for such operation will increase in proportion to the frequency (provided a system were available to require

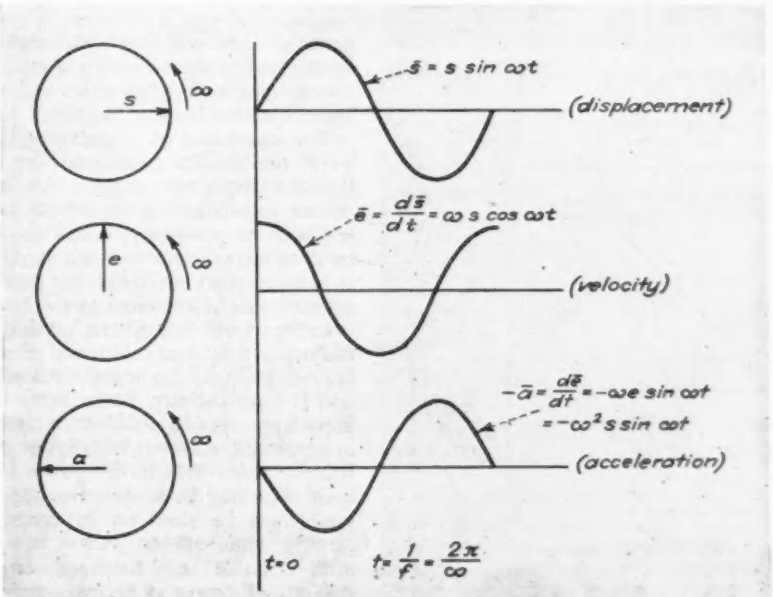


Fig. 5—Relationship between displacement, velocity and acceleration of a sinusoidally oscillating point as to magnitude and phase. Displacement amplitude is the most tangible measure of armature or plunger movement, but the velocity amplitude is more useful. Because of variations with time, units of measurement must be fully checked.

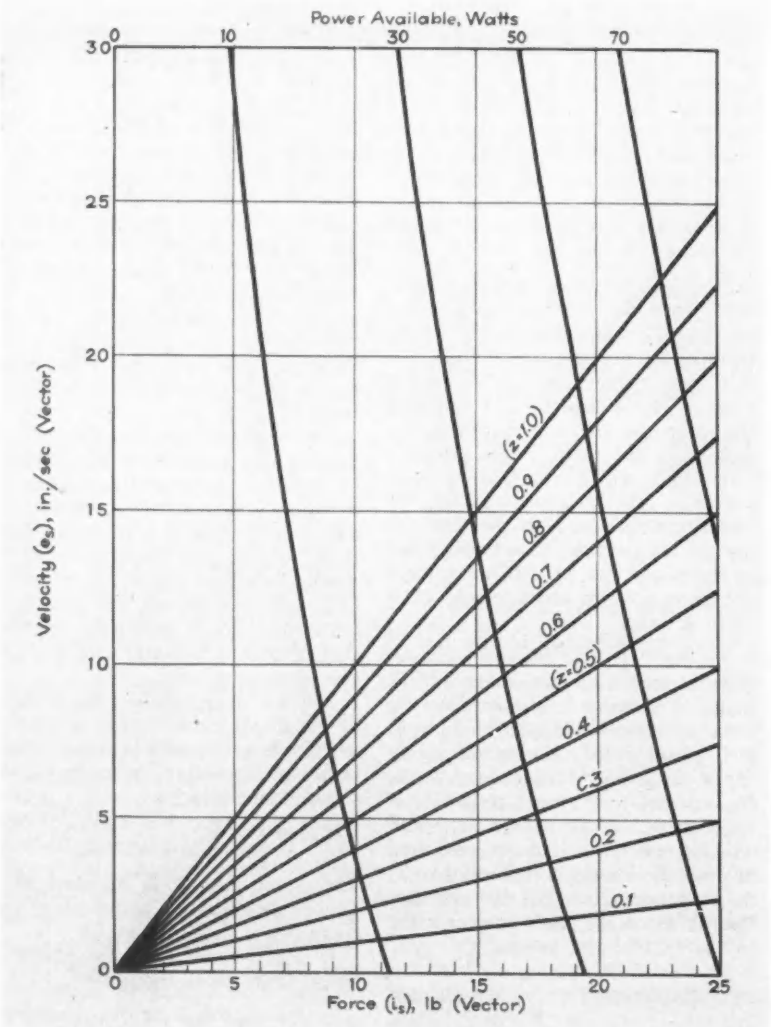


Fig. 6—Performance data for a single shaker operating into resonant mechanical load.

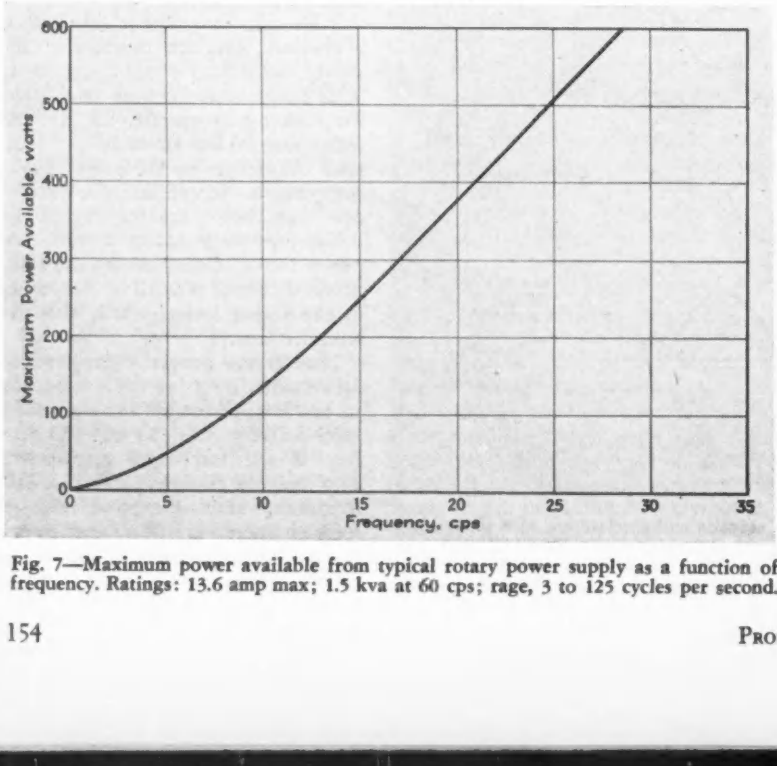


Fig. 7—Maximum power available from typical rotary power supply as a function of frequency. Ratings: 13.6 amp max; 1.5 kva at 60 cps; rage, 3 to 125 cycles per second.

this force-velocity combination).

Stated differently, a limitation of stroke is imposed by the armature design, a limitation of force by the overall shaker design and a limitation of power available by the power supply.

To summarize, in estimating shaker performance, it is necessary to specify the following relationships:

(1) The type of job the shaker will be required to perform: From the category of uses described previously, the vibration engineer or shaker designer is able to define the character of the mechanical load. This is conveniently done in terms of the mechanical mobility, which, by analogy, is similar to an electrical impedance, and can be handled as such when designing the power equipment to operate the shaker over a wide frequency range. In the case of resonance testing, the equivalent reflected impedance of the mechanical system is resistive, while for "brute force" shaking of masses, it is a capacitive reactance.

(2) The electrical impedance characteristics of the shaker: These include the resistance of the coil, coil inductive reactance which goes up with frequency, and the equivalent resistance caused by hysteresis and eddy current losses in the pole pieces, which are also a function of frequency.

(3) The power available from the power supply as a function of frequency: Chart in Fig. 6 shows the interrelation of the first two variables for a particular shaker when used for resonance tests. The unit is limited in its direct use to a maximum frequency range of 500 cps. Curves of constant power available are shown with shaker vector velocity as ordinate and shaker vector force as abscissa. The sloping lines represent constant mobility.

The chart in Fig. 7 indicates the third relationship mentioned for a typical rotary power supply. An important requirement is the control system between power supply and shaker to allow power factor correction and impedance matching so that the power available can be delivered to the load.

In the case of "brute force" shaking, endurance testing or calibration of vibration pickups, the main consideration is force delivery, power being of importance only in sufficient quantity to supply the relatively high losses involved. Even though the power requirements may be modest, the equipment must be sized on its current-carrying requirements. Power equipment capable of handling large amounts of power at ordinary power frequencies must be used at much reduced ratings in order that the same currents can be delivered at much lower frequencies.

Solving the

Pythagorean Principle

On the Slide Rule

HENRY W. D'ENTREMONT

IN EVERY BRANCH OF ENGINEERING, there are problems that call for the use of Pythagoras' principle in their solution. A resultant force, an impedance or a simple dimension must be figured. Being unaware of the slide rule, Pythagoras did not consider making his relationship adaptable to two or three simple mechanical operations. As a result, much time is consumed in looking up or calculating squares and square roots.

If a slide rule solution to a problem is sufficiently accurate, there is a short-cut method that is well adapted for use on this instrument. Once memorized, the steps are quickly accomplished.

For a right triangle of base g, altitude h and hypotenuse f

$$f = \sqrt{g^2 + h^2} = h\sqrt{\frac{g^2 + h^2}{h^2}}$$

$$= h\sqrt{\left(\frac{g}{h}\right)^2 + 1}$$

In all cases, the larger numerical value can be assigned as g, so that g divided by h is greater than unity.

Referring to Fig. 1(A), h is set on the C-scale over g on the D-scale. The value of $(g/h)^2$ is then read on the A-scale over the left hand index of the B-scale. The factor *one* (under the radical in the last equation above) is then *added* to the value just read, and the B-scale index reset to the increased value as shown in Fig. 1(B). Finally, the desired value of f is found under h on the C-scale.

If the values of the hypotenuse and either leg of the triangle are known,

$$h^2 = f^2 - g^2 \text{ or } h = g\sqrt{\left(\frac{f}{g}\right)^2 - 1}$$

The procedure is as above except that the factor *one* (under the radical) is *subtracted* from the value found on the A-scale and h is found under g on the D-scale.

The main precaution to observe is that if the value of $(f/g)^2$ is between *one* and *two*, this value minus *one* is a decimal fraction less than *one*. In this case, the right hand index on the B-scale is set under the decimal as shown in Fig. 1(C). Then, under the value of the known leg on the C-scale, the value of the unknown leg is read on the D-scale.

If the ratio of g to h is high or if the ratio of the hypotenuse to one leg is low, care must be taken to be sure that correct scale-settings are made.

Large numerical values, such as those for forces, should be reduced by moving the decimal point to the left to obtain a value between *one* and *ten*. The decimal point must, of course, be restored the same number of decimal places in the result. Likewise, small figures less than one, should be increased to place the smaller value between *one* and *ten* and then reduced the same number of places in the result.

Sample Calculations

Problem 1. The right-angle components of a force are 16,500 and 9,200 lb. Problem is to find resultant force.

Solution: Moving the decimal points to the left, 9.2 is set on the C-scale over 16.5 on the D-scale. 3.22 is read on the A-scale over the right hand index of the B-scale. This index is then moved up to 4.22 on the A-scale and 18.9 found on the D-scale under 9.2 on the C-scale. Replacing the decimal point, resultant is 18,900 lb.

Problem 2. It is desired to figure the reactance of an electrical circuit containing an impedance of 69 ohms and a resistance of 16 ohms.

Solution: Since the hypotenuse (69) and one leg (16) are known, 6.9 is set on the D-scale opposite 1.6 on the C-scale. As in Problem 1, 18.6 is read on the A-scale, except that the *left hand* index is used. *One* is subtracted from this last value because one leg of the triangle is the unknown. The answer is found by reading 6.7 or 67 ohms under 1.6 set on the C-scale.

Problem 3. A resultant velocity of 590 fpm and one right-angle component of 462 fpm are known. The other component velocity is to be figured.

Solution: Again, the hypotenuse and one leg are given. 5.9 and 4.62 are set on the D- and C-scales and the A-scale value read as 1.63. Subtracting *one* and resetting the *right hand* index to 0.63, 3.67 is found opposite 4.62 on the C-scale. The answer is therefore 367 ft per minute.

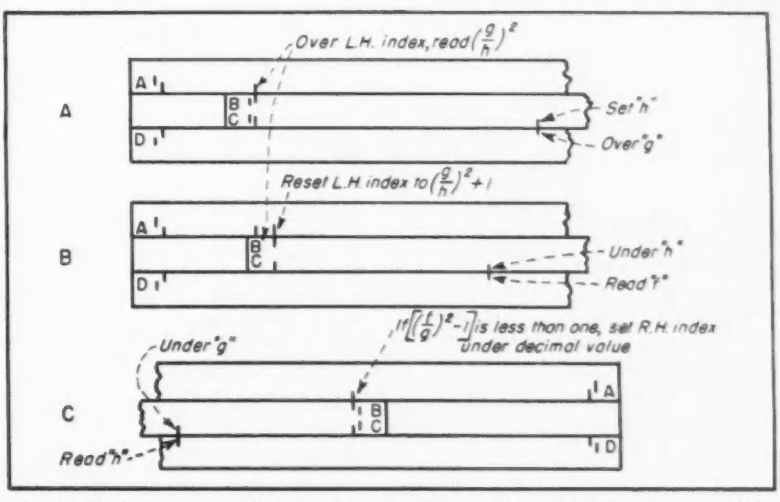
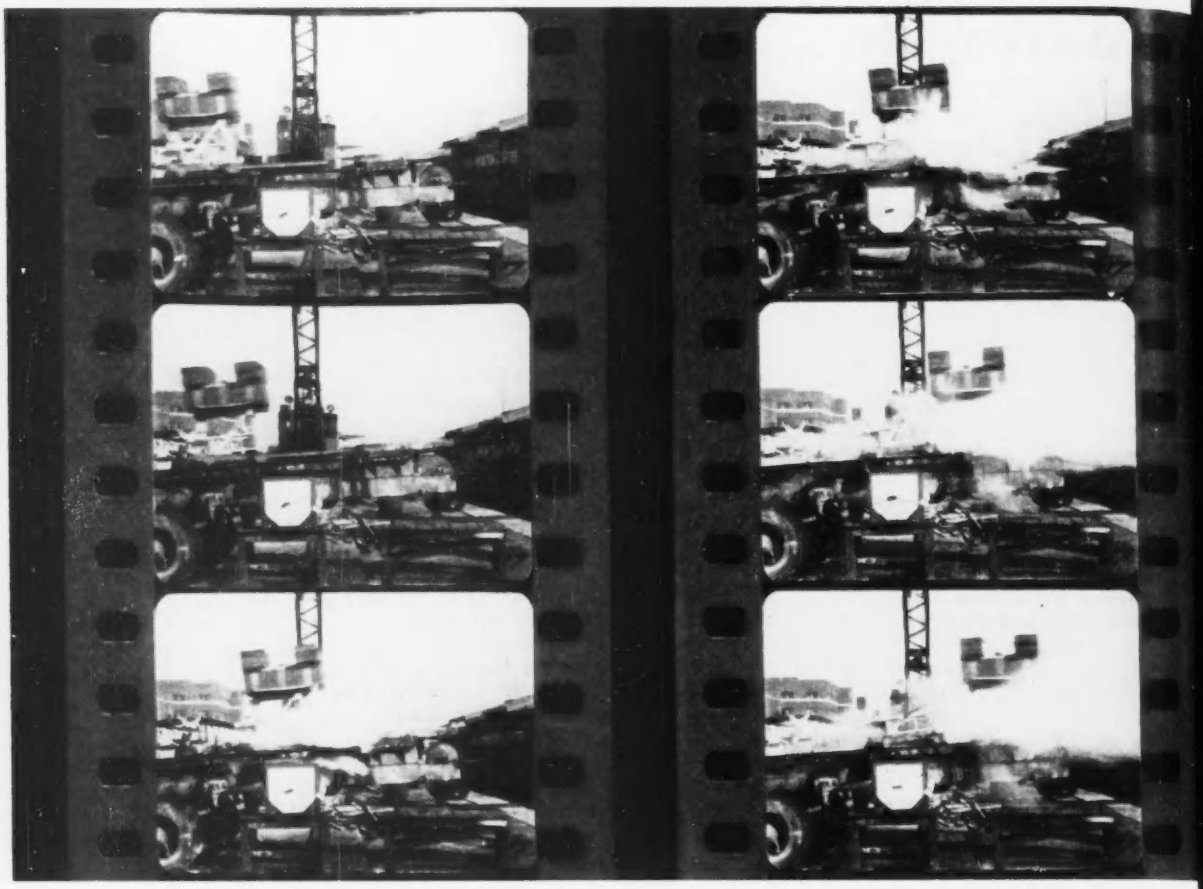


Fig. 1—Typical settings of slide rule scales for solution of Pythagorean principle. (A) and (B) are for figuring the hypotenuse; (C) for calculating an unknown leg.



FILM STRIP—Picture sequence on these pages is from a high speed film of an airplane catapult test run. In the first picture, the catapult car, carrying a dummy load, is seen approaching the water brake at 70 mph speed. After the piston ruptures the brake

Selection Factors for

Various buffer types and the phenomena that influence their operation. Advantages and limitations of each buffer type are listed. First of two articles on this subject by the author.

JOSEPH BROWN*
Engineer, Naval Aircraft Factory
Philadelphia, Pa.

BASICALLY, A MECHANICAL BUFFER is a device that applies a retarding force to a moving body so as to safely bring the body to a halt in a predetermined distance. Any brake, shock absorber, or dash pot, may be so classified. The term mechanical, however, is used in the broad sense since it includes types in which the energy of deceleration is absorbed by mechanical, hydraulic or pneumatic means. Electric and magnetic buffers also are practical, although they do not as yet have the universal application of the mechanical type.

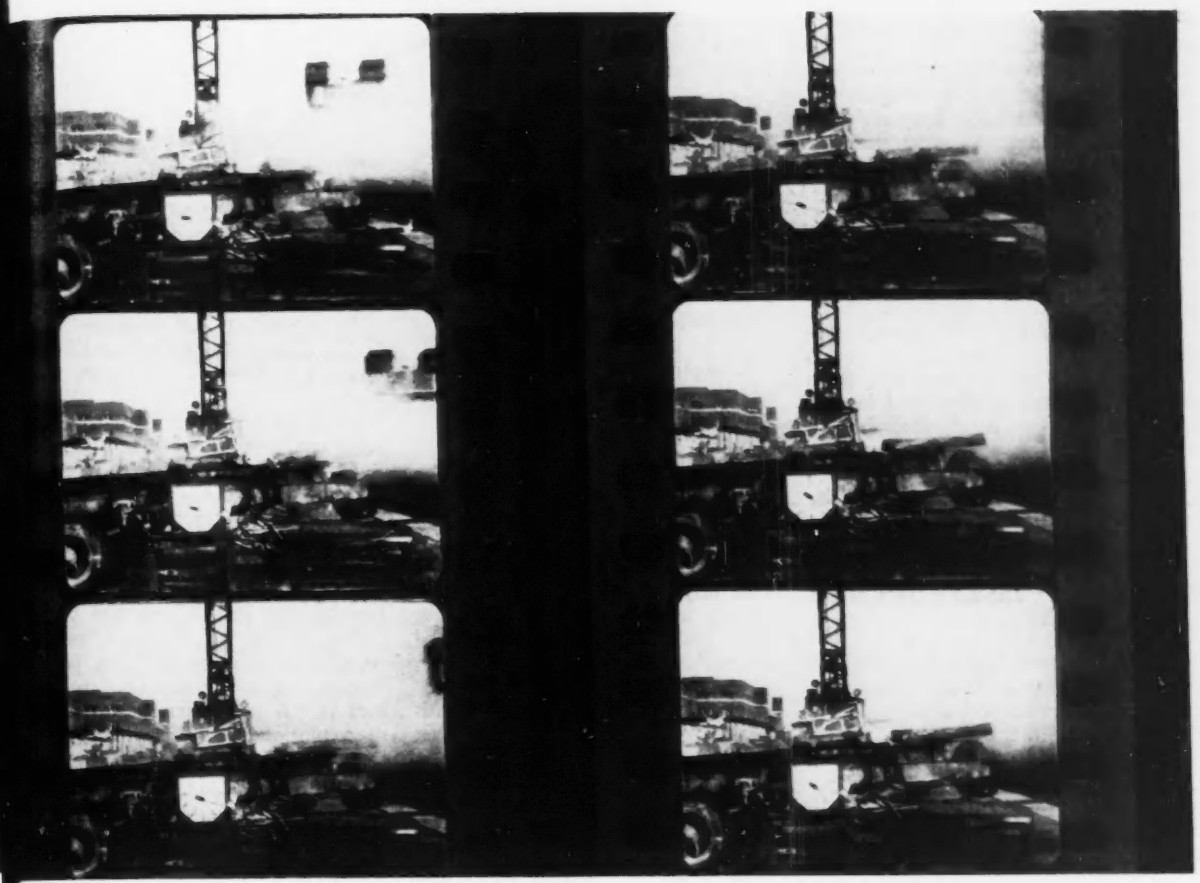
In general, mechanical buffers are either one or a combination of two basic types. One type of buffer converts the kinetic energy of the moving body into heat; friction brakes, water brakes and hydraulic shock absorbers are examples. The second type of mechanical buffer does not convert kinetic energy, but stores it and returns most of it to the body being stopped. The air brake, Dowty brake, springs and bumper pads are in this classification.

The best type of buffer for any particular need depends on such factors as speed of the moving body, time between successive buffer applications, expense of manufacture, simplicity of the mechanism, and space available for installation. A brief description of the various buffer types, together with suitable comments concerning their characteristics, follows.

Orifice Type Buffers

These devices transfer the kinetic energy of the moving body to a buffer fluid, where the energy is transformed

* *The opinions and assertations contained herein are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Dept. or the Naval Service at large.*



diaphragm, the car is brought to a halt within two ft, while the dummy load continues on. One revolution of the clock hand represents one sec of time. Both the water and the diaphragm used in the brake must be replaced after each operation.

Mechanical Buffers

into heat. The fluid is generally a liquid of low viscosity, or less often, a gas. Hydraulic buffers and water brakes are the chief examples.

HYDRAULIC BUFFER. The sketch in Fig. 1 shows the essential operating parts of a hydraulic shock absorber. This type may be used in applications where the speed is below 40 mph. Bodies moving faster than this limit set up excessively high stresses on the piston. The compressive stress on a steel piston is in the order of 100,000 psi when it is struck by a body traveling at 38 mph. However, when compound buffers are used, speeds up to 70 mph are allowable.

Note that speed—not the combination of speed and weight—is the principle factor in establishing buffer ratings. Buffer impact pressures are affected only by the speed of the body to be stopped. The stresses in the piston or fluid at impact are determined by the speed of impact, and it is these stresses that limit buffer capacity. Weigh enters the picture only after impact occurs. The physical size of the buffer and the strength of the cylinder walls determine the maximum weight that can be stopped safely. Thus, increasing buffer size or "beefing up" cylinder walls makes possible the handling of heavier bodies. But once the speed limit is exceeded, the piston material deforms upon impact. Changes in buffer size or wall strength are to no avail. Hydraulic buffers have the following advantages.

1. Dependable operation.
2. Little maintenance required.
3. Compactness and lightweight.
4. Long service life.
5. All absorbed energy is dissipated as heat.
6. No need for auxiliary apparatus, such as high pressure air supplies.

The main disadvantages are that they are moderately expensive, and the maximum speed at buffing must be below 40 mph.

COMPOUND BUFFERS consist of two hydraulic buffers acting in series. One buffer is mounted on the moving body and the other on a stationary structure. The arrangement illustrated in Fig. 2 is typical. The additional buffer has the effect of reducing fluid impact pressures and piston stresses. Hence, greater loads can be accommodated. Otherwise the advantages and disad-

vantages are similar to those of the simple hydraulic buffer described above. This type buffer has been successfully used to absorb the energy of a 1,200 pound body traveling at a speed of 70 mph, in a 30 in. brake stroke.

WATER BRAKE. The speed limitation of the hydraulic buffers is avoided in the water brake. In contrast to the hydraulic buffer, the piston in a water brake is part of the body to be stopped. The fluid, which is expendable, is water in most cases and is usually retained in the brake by a thin diaphragm, as illustrated in Fig. 3. The energy is absorbed by the piston piercing the diaphragm, entering the water chamber and forcing the water through the clearance between the piston and cylinder. The diaphragm, if one is used, and water must be replaced after each operation of the buffer. The advantages of the water brake are:

1. Economy of construction.
2. Ability to withstand impact at high initial piston velocity.
3. Dependable operation.
4. Compactness and lightweight.
5. Complete dissipation of absorbed energy.

The disadvantages are:

1. Provisions must be made for fast replacement of water and diaphragms especially where successive applications are in rapid order.
2. Piston weight usually adds considerable inertia to moving body.
3. Suitable provisions must be incorporated if cold weather operations are to be conducted.
4. Liquid ejected from the brake may become an inconvenience and corrode surrounding equipment.

Compression Type Buffers

Compression type buffers transfer the energy to be absorbed to the brake medium, where it is stored. This energy must be dissipated before the brake may be used again. The brake medium may consist of spring coils, rubber pads, or gas or liquid confined in a suitable vessel.

SOLID SPRINGS. Included under this classification are coil springs, rubber or synthetic solid material pads or any other type of solid elastic medium. The moving body is brought to rest by striking the buffer and transferring its kinetic energy to the buffer in the form of internal energy of compression. Solid springs are simply constructed, require little maintenance and have high endurance in low speed applications. However, impact shock is severe, restricting designs to very low speeds. Also, these buffers are large in relation to the amount of energy absorbed. A third limitation is that energy is not dissipated but is returned to the mass being decelerated.

SIMPLE PNEUMATIC. Basically, this brake consists of a piston and cylinder. Buffing is accomplished by the piston covering the exhaust port and compressing the air trapped in front of it. The initial air pressure within the brake cylinder is atmospheric. The air is compressed to a very high pressure in most cases at the end of the buffing stroke. As a result there is considerable leakage past the piston, despite elaborate precautions to prevent it. A more serious problem is the explosion of lubricating oil within the cylinder as a result of the high compression.

The advantages of this brake are:

1. It is possible to use the stored energy in the compressed air to return the piston to its original position.
2. Construction is inexpensive.
3. Endurance is good.
4. Maintenance is low.
5. Size and weight are reasonably small although larger than for the water brake.

The disadvantages of this brake are:

1. Most of the brake stroke is used in building up pressure.
2. Air leakage is difficult to over-

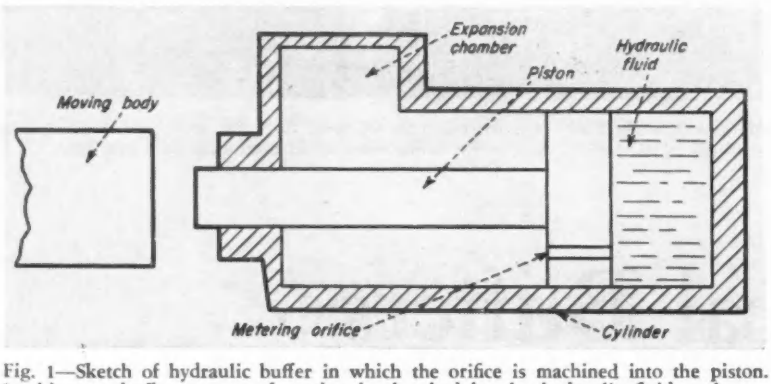


Fig. 1—Sketch of hydraulic buffer in which the orifice is machined into the piston. In this type buffer, energy of motion is absorbed by the hydraulic fluid as heat.

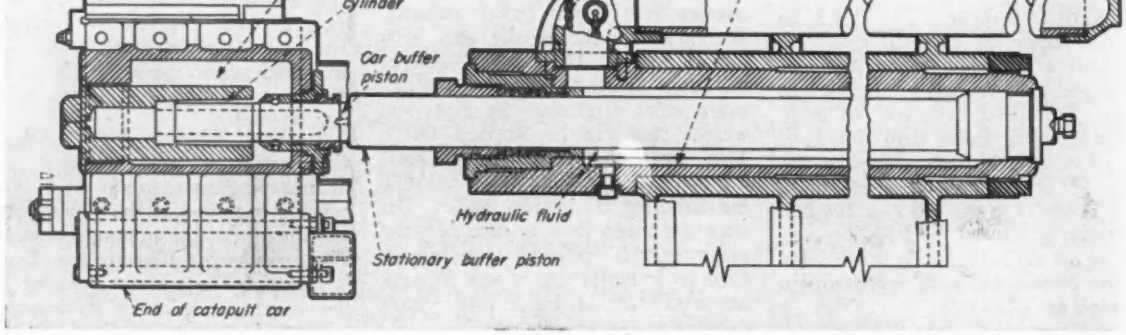


Fig. 2—Compound hydraulic buffer is used to hold the car of an airplane catapult. One buffer is stationary while the other is mounted on the catapult car. The 1,200 lb car is moving at 70 mph at the instant of impact.

come, thereby resulting in a larger buffer stroke and thus a heavier brake cylinder and longer piston. A long buffer stroke has the advantage of reducing stresses in the moving body by decreasing its rate of deceleration. A short stroke, of course, allows use of a correspondingly short—and hence lightweight—buffer cylinder. Which alternate is chosen depends on the individual design limitations involved in the application.

3. Provision must be incoporated in many cases to prevent piston rebound at excessive speed.

4. Tendency of trapped oil or grease to burn as a result of the diesel effect.

5. Peak pressure to average pressure ratio is usually very high.

BY-PASS PNEUMATIC. The simple pneumatic buffer uses most of the brake stroke in increasing atmospheric pressure to one high enough to produce a suitable retarding effect. There are means employed to increase this initial pressure, thus considerably improving the buffing efficiency. The provisions shown in Fig. 4 may be applied where the moving body does not act directly upon the buffer piston but rather through a cushion of compressed air. As the piston uncovers port A, air from the driving side of the piston is allowed to flow into the brake chamber. This flow stops when the piston covers port B and braking begins. The capacity of this type brake is many times that of the simple pneumatic buffer. Otherwise the same advantages and disadvantages apply.

PNEUMATIC BUFFER WITH AUXILIARY HIGH PRESSURE SUPPLY. Here a different means is used to pressurize the buffer cylinder prior to the start of the brake stroke. This is usually a source of high pressure gas. The disadvantages of this brake lie in the greatly increased complexity of design. Means must be incorporated to admit the high pressure air at the suitable time and to shut the supply off to avoid loss of the high pressure gas. Large size valves must be used to admit the correct weight of air in the short time provided.

DOWTY BRAKE. With this device, Fig. 5, it is possible to obtain retarding action in both directions of piston travel. Check balls in the piston head open the ports (see Fig. 5) during the impact stroke. These ports close during the return stroke, leaving only several small orifices open to fluid flow. Thus, during the impact stroke the device is a liquid spring, while during the return stroke it behaves as a hydraulic buffer. Analytical determination of return orifice size is treated later.

The advantage of this brake lies in its compactness of construction. However, the brake stroke is very small. A speed of about 40 mph is the maximum that can be handled by the Dowty brake.

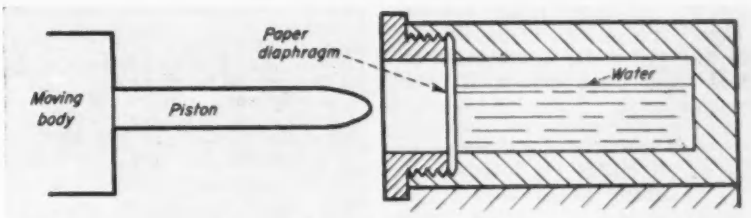


Fig. 3—Schematic of water brake. Both water and diaphragm are expendable.

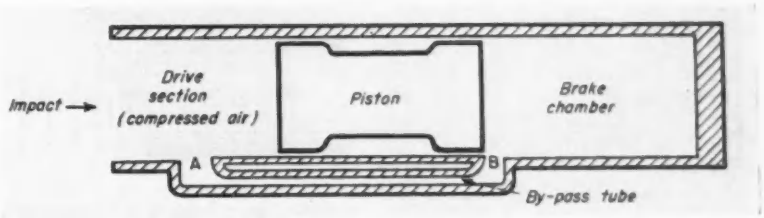


Fig. 4—In the by-pass pneumatic buffer sketched above the brake chamber is supercharged by air from the drive side. This reduces the distance that the piston must travel before effective braking begins and permits use of a shorter brake chamber.

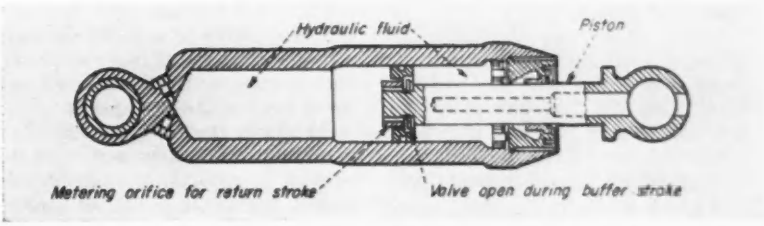


Fig. 5—Piston of Dowty brake is equipped with ball check valves. These are open during impact stroke, but closed during the return. Thus, the device is double-acting, with greater resistance on the return stroke. Suitable for speeds up to 40 miles per hr.

Brake Combinations

There are innumerable buffer combinations that can be used to obtain particular effects and to satisfy practically any design need. The sketch of Fig. 6 illustrates the combination of a water brake with a pneumatic brake. The purpose of the water brake is to accelerate the pneumatic brake piston with a minimum of impact. The greater part of energy is absorbed in the air brake. The hydraulic fluid in the air brake is merely a connecting link between the air mass and the piston. The air mass is at high initial pressure, thus avoiding the serious objection of the simple pneumatic brake.

To sum up, the hydraulic buffer is the most satisfactory for low speed operation (up to about 25 mph). It is also recommended for uses where large weights and kinetic energy are involved. This type will require no auxiliary apparatus, will provide trouble free operation and will have fairly constant deceleration characteristics.

For low speed operation where weights are relatively light and kinetic energy is small, a simple pneumatic cylinder will be least expensive. Rebound of the body after buffing must be tolerated. If this is undesirable, it is preferable to use a hydraulic buffer instead of complicating the mechanism to avoid rebound.

For high speed experimental operation, where kinetic energy is considerable and where rapid consecutive operations are not required, the least expensive buffer will be the water brake. The amount of excess power is determined by the additional energy input required for the water brake piston that is carried on the moving body. Impact effects upon the body may be reduced to a minimum by suitable design of spear entrance contour, although pressures will not be affected perceptably.

The pneumatic buffer, with auxiliary high pressure gas supply may also be

adapted to high speed operation. There are a number of disadvantages to this type buffer as compared to a high speed water brake which should be considered.

Theoretical Treatment of Hydraulic Buffers During Initial Impact

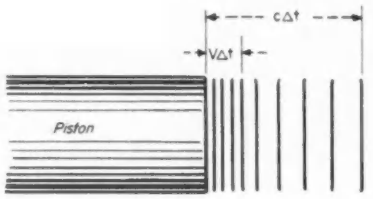
Prior to discussing the phenomena occurring during actual buffing, it is desirable to analyze the instantaneous conditions as they exist at initial impact. Two previous articles by the author ("Hydraulic Shock Absorber Orifice Design—Parts I and II", PRODUCT ENGINEERING, February and April 1948) neglected these conditions since the buffers discussed were of the low velocity type. In higher speed brakes, however, the preliminary effects of impact must be considered. Impact conditions affect all buffers containing a liquid buffing medium. They also affect gaseous brakes at supersonic velocity.

Three phenomena determine the pressure resulting from impact between the piston and the hydraulic fluid. One, the fluid undergoes compression during its acceleration to piston velocity. Two, the orifice restricts fluid flow, thus increasing pressure within the brake cylinder. Three, the inherent flexibility of the structure bearing the piston and the compressibility of the piston material tends to dampen impact pressure peaks.

Fluid compressibility is important in higher speed brakes. Its determination is very complex involving pressure waves which continuously reflect from the cylinder walls and the back of the orifice flange. Nevertheless, it is possible to closely predict the maximum liquid pressure which will result for a given speed. Let

V = body speed
B = Bulk Modulus of the liquid
c = speed of sound in liquid
v = volume

From the sketch it is seen that in a small interval Δt, the space affected by the impact will extend $c\Delta t$ from the piston. During this same time



interval, the piston will have traversed a space $V\Delta t$. From the definition of Bulk Modulus:

$$\Delta p = -B\frac{\Delta v}{v} = -B\frac{V\Delta t}{c\Delta t} = \frac{-BV}{C} \quad (1)$$

The negative sign of the derivitive dp/dv indicates a negative slope, i.e., as p increases v decreases. Thus, if Δp is positive, Δv is negative.

Peculiarly, the nose contour of the impacting piston does not affect the pressure at impact. Neither does the relative size of the piston with respect to cylinder diameter materially affect this pressure. The elasticity of the piston does slightly reduce the pressure. Experiment has shown that the measured pressure will generally be about 80 percent of that computed from the above formula.

The buffer designer can allow for impact by computing the impact pressure and substituting this value in the formulas that assume impactless buffing. This result will be a buffer of slightly longer stroke than necessary, which is on the safe side.

The two other considerations are somewhat less important. The orifices for initial buffing are usually too large to materially increase retarding pressure, and hence can be neglected. Except in the case of the compound buffer, the elasticity in buffer bearing structures is generally low and can be overlooked for conservative design.

It sometimes happens, however, that where impact conditions are critical and the structure is flexible, advantage can be taken of the shock reduction induced. In such cases this procedure may be followed. From Castigliano's theorem can be obtained the structure deflection at the point of buffing for one lb of buffing force. The first step is to find the equivalent mass for the structure at the point of contact with the buffer.

Then if F = buffer force,

Fg/W = structure acceleration = f

F also equals $K\delta$ where δ is the deflection and K is a constant.

Thus $K\delta g/w = f = vdv/d$ since $fd\delta = vdv$ from which $v^2 = k\delta^2\, g/w$ and $v = \sqrt{k\delta^2\, g/W}$

This velocity reduces the impact velocity, thus lessening impact pressure, which is obtained from Eq (1).

In a second article, the author amplifies the discussion of buffer selection. Design considerations and formulas will be given for constant deceleration buffers, compound buffers, compression type buffers and others. In most cases, the formulas will simply be stated and advice given as to their use. But formulas for compound buffers and constant deceleration brakes will be derived step by step.

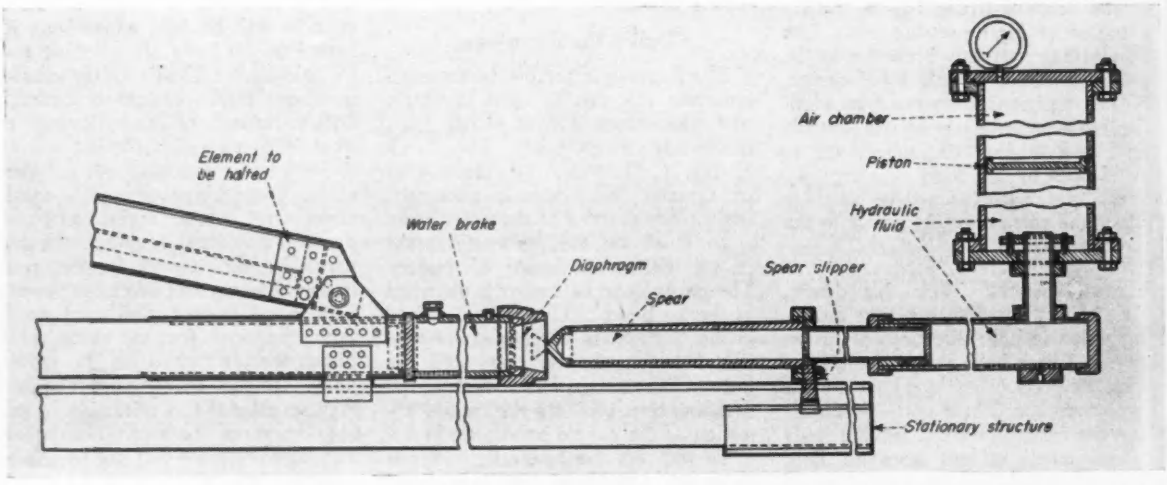


Fig. 6—Combination buffer comprising a water brake in series with a pneumatic brake. The latter is a special design with hydraulic fluid between the spear and the piston. Air can be trapped at a high initial pressure to produce immediate braking effect.

TECHNICAL NEWS

World's Longest Milling Machine is Unveiled

ONE OF THE SPECIAL PRODUCTION MACHINE TOOLS to produce advanced types of aircraft is the new Onsrud Spar Mill, recently installed in Boeing Airplane Company's Seattle plant. With an over-all length of 105 feet, the machine is believed to be the longest milling machine ever built. It will be used to production-machine a variety of parts.

Airplane spars are tough aluminum extrusions usually 24 ST or 75 ST alloy. The spars require many different types of milling operations such as tapering cuts in the webs and flanges and twist cuts along the length of the spar. The mill permits milling the entire length of such a part and the making of any types of milling cut required during one pass of the carriage. In addition, since the positions of the cutterheads are controlled by the pattern or template method, milling cuts may be made involving non-mathematical curves as required by design, typical of many of the parts that must be made in aircraft production.

The carriage of the milling machine houses four cutter motors with a total of 200 hp available. Two of these motors are in a horizontal position and two are in a vertical position. The carriage travels over the milling machine bed, from one end to the other, driven by a d-c motor and spiral tooth rack assembly under electronic control. Speed of carriage travel is infinitely variable from 10 to 200 inches per minute and may be set to vary automatically in correct relation to the work being milled.

Many types of cuts that can be made on the mill are possible only because of the design of the cutter motor mounting assemblies. The vertical motors are mounted on parallel arms and can slide up and down and can also be tilted to various angles. Each motor can be guided for position separately under template control as the carriage moves forward in feed. As determined by the Template, the motor raises or lowers or tilts, to mill the particular face desired on the work at any given point throughout the length of the part.

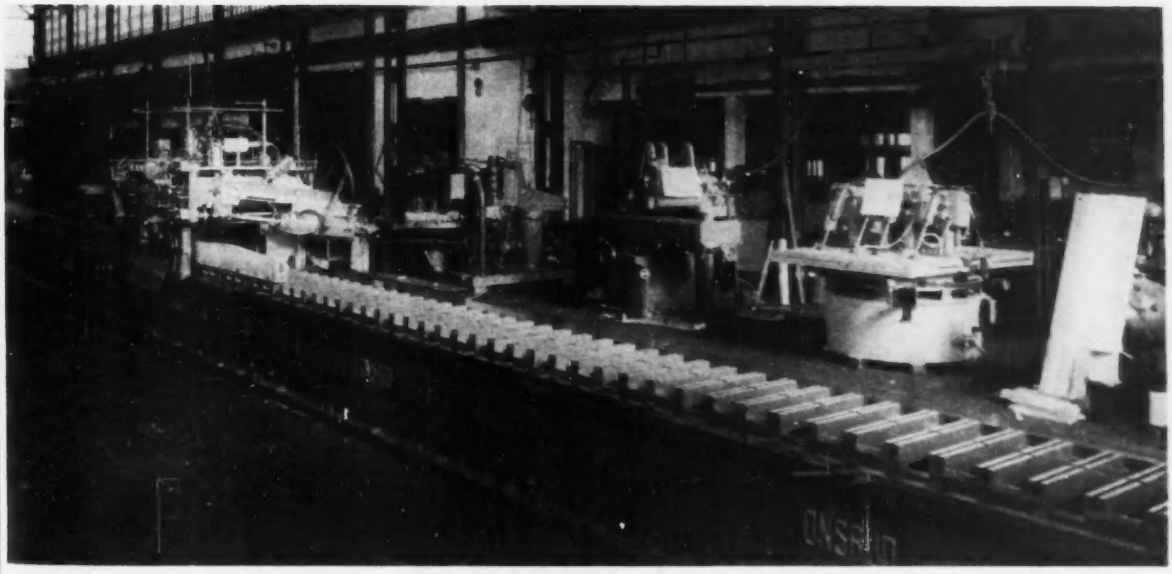
Similarly, the horizontal motors raise and lower and tilt, to assume new cutting positions for the cutters as the carriage travels along the bed. The vertical motors are high frequency electric and the horizontal motors operate on standard frequency. Both types of motors are liquid cooled. Template followers and cutters motors are held to the patterns and the work pneumatically.

The cutters used are carbide tipped. Metal is milled and removed at the rate of 2½ to 4 cubic inches per minute per hp for a total of up to 800 cubic inches per minute. Cutter motors and cutters are designed to provide a peripheral speed average of 7,500 feet per minute. Coolant is supplied to all cutters, and is collected in the bed, filtered and recirculated by an 80 gpm centrifugal pump.

Basic specifications include: over-all length, 105 ft; bed length, 90 ft; maximum carriage travel, 80 ft; weight, approximately, 120,000 lb.

International Patents' Institute

THE HAGUE—The International Patent Institute (Institute International des Brevets), whose members are France, Belgium, the Netherlands and Luxembourg, has been officially inaugurated at the Pace Palace in The Hague. The Institute was created by a treaty signed by the four countries in



OVERALL LENGTH OF THE MILLING MACHINE is 105 feet. The bed is so long that it is difficult to show the entire machine in one illustration. Carriage is in approximate center position with short sample work set-up on bed.

1947 and ratified in 1949. It has a staff of about 25 technical experts of various nationalities and an executive committee including one delegate from each member country.

The task of the new Institute will be to ascertain whether an invention is really new, but it will act only in an advisory capacity. In Holland, Britain and America a patent is granted only after official investigations proving that an invention is entirely new. In other countries, such as France, Belgium and Luxembourg, patents are granted automatically and patent holders are often summoned for contravention.

Airplane Model Profiles Tested in Water Channel

DEVELOPMENT OF AN ELECTRONIC DEPTH INDICATING PROBE MECHANISM has enabled Georgia Institute of Technology, Atlanta, to obtain a high degree of accuracy in testing aircraft model profiles in its newly constructed water channel. The electronic device enables accuracy of measuring the local water depth along the model profile to 1/1,000 in. Water depth in the channel itself is measured to accuracy of 1/10,000 inch.

This unique water channel, developed by H. W. S. La Vier, research associate engineer, is providing means of testing aircraft components in the supersonic, transonic and subsonic speed ranges without incurring the adverse shock wave "choking" so common in normal wind tunnels.

The shallow channel water table at Georgia Tech is 20 feet long and 4 feet wide with water completely covering the plate-glass bottom at a quarter inch depth. Models are towed through this shallow water and the waves caused by the model and the distribution of the water depth about the model provide data to which hydraulic analogy is applied.

This analogy is based on the similarity of the wave propagation velocity for a shallow channel of water with a free surface and the pressure propagation velocity of a compressible gas.

The ratio of local pressure at any point on a profile to the free stream gas pressure is analogous to the square of the local water depth at the same point on the similar water channel profile divided by the undisturbed channel depth. The analogous relations are used to convert the observed water channel depth data into pressure and velocity distribution data similar to that resulting from tests conducted in supersonic wind tunnels.

Models being studied are towed back and forth throughout the length of the channel by a carriage driven by an endless cable system. Wave formations about the model are photographed and an electronic probe system has been devised to measure the local water depths around the profile.

Wave propagation velocity for shallow water with a free surface quarter inch deep is 0.82 feet per second. This is directly analogous to wave or pressure propagation velocity of a compressible gas such as air. Sonic velocity of standard air is 1088 feet per second.

By varying the speed of the model any Mach number can be obtained. The Georgia Tech water channel is geared so that a range of testing speeds can be obtained from half to as high as seven times the speed of sound. Variation of the carriage and model speed is accomplished by adjustment of a vari-speed drive mechanism with power from a one-quarter horsepower electric motor.

The electronic probes are so placed that as the model passes through the test field each probe is adjusted until it contacts the water. At the instant of contact a light flashes. Each probe is adjusted to the wave formation and from these distances the depth along the model profile is obtained. The wave formations are the same as air waves when the aircraft is in flight.

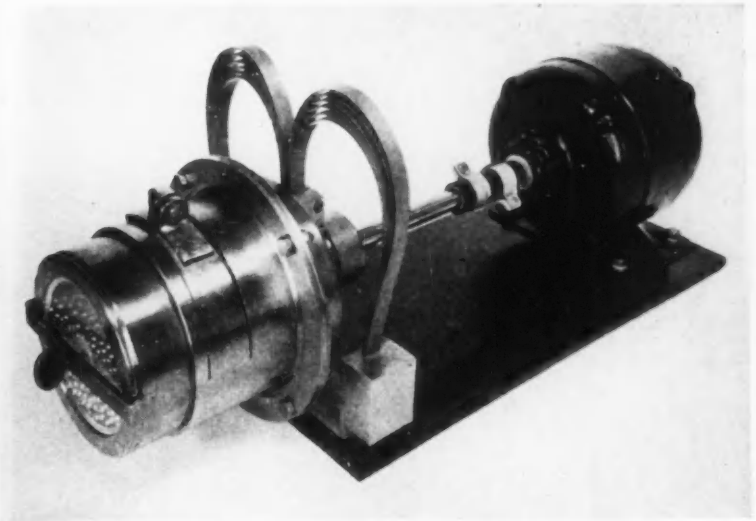
Big advantage in the use of the water channel is that any desired speed can be obtained and tests can be made quickly and inexpensively.

Value of the Georgia Tech water channel was recently illustrated in its own aerodynamic laboratory where trouble was being experienced in the newly constructed small supersonic wind tunnel. A model of the throat of the tunnel was made and tested in the water channel. They revealed irregularity in the walls and throat as causing objectionable shock reflections off the tunnel walls. The tunnel was modified and excellent results are being obtained.

Another great value of the water channel is that during the designing of supersonic aircraft each component part can be tested and either changes suggested or the idea junked. This would not be possible in other than a supersonic wind tunnel.

ASME Rubber and Plastics Division Meets

THE RUBBER AND PLASTICS DIVISION OF THE ASME will meet at the Hotel Statler in New York on November 30 and December 1. The morning sessions will be given over to discussions of applications of rubber, plastics and adhesives, three papers being presented each day. The afternoon of November 30 will be given over to a symposium on gaskets and packings, held jointly with the ASME Machine Design Division.



ASSEMBLED VIEW. This vibratory ball mill recently constructed at the National Bureau of Standards is expected to prove useful in the grinding and blending of materials. Grinding takes place between the colliding steel balls in the jar at left.

Vibratory Ball Mill for More Efficient Grinding

A VIBRATORY BALL MILL recently developed at the National Bureau of Standards rapidly and completely grinds cotton into particles a few microns in length with little or no oxidation change or contamination of the cellulose. Grinding takes place as 3,700 steel balls, colliding over 100,000 times per second, pulverize the material caught between them. As the new mill produces a more uniform powder and is more efficient and easier to operate than previous devices of this kind, it is expected to find extensive application in the grinding and blending of a variety of substances such as pigments, ceramic materials, metal powders, resins, and plant and animal tissues.

The mill was constructed by Florence Forziati, W. K. Stone, J. W. Rowen, and W. D. Appel of the Bureau staff in connection with a program sponsored by the U. S. Department of Agriculture on the chemical deterioration of cotton during processing and use. To learn more about the chemical changes which occur when cotton deteriorates, the Bureau undertook to apply the techniques of infrared spectro-photometry to cotton cellulose. As the spectra obtained from fibers laid side by side were not satisfactory, it was decided to reduce the fibers to a powder and measure the spectral absorption of the powder in a mat or in suspension in a suitable liquid. Several well-known methods for grinding materials were therefore tried, such as an ultrasonic generator and a Wiley mill in which the material is cut between fixed and moving blades. However, the products thus obtained contained much coarse matter. In an effort to provide a method which would reduce the entire sample to a finely divided state without contamination and with a minimum of chemical change, the vibratory ball mill was developed.

In the Bureau's mill, a cylindrical jar suspended by leaf springs and containing steel balls in contact with the material to be ground is swung through a circular path. This excites the steel balls, causing the numerous collisions that grind the materials in the jar. At the same time the jar rotates slowly about its axis. In this way, the material being ground is prevented from settling to the bottom of the jar, and uniform grinding is obtained.

Motion of the jar is produced by means of a horizontal shaft connected at one end, through a flexible coupling, to a 0.25 hp motor and at the other end to an eccentric weight. The weight, rotating at a speed of 1,800 rpm, drives the weight housing in a circular path that is confined within certain limits by the flexural resistance of the springs. The jar containing the sample is held in a cylindrical holder, which is attached to the weight housing by means of a greased slip ring. A small difference in diameter between the flange of the jar holder and the slip ring groove permits the holder with the clamped-in jar to be rolled around the groove circumference by centrifugal force. This rolling or rotation of the holder and jar, which takes place at a rate of about 4 rpm in a direction opposite to that of the rotating weight, continually redistributes the material being ground.

In experiments at the National Bureau of Standards, the mill reduced 5 grams of cotton in 30 minutes to particles 10 microns or less in major dimension. X-ray diffraction measurements showed that the cellulose was converted almost completely to the amorphous form in this time. Except for the decrease in degree of polymerization which resulted from the cutting of the cellulose chains, there was no degradation of the cotton. Nor was the ash content increased.

U. S. Tractors Are Long Lived

DESIGNERS OF FARM MACHINERY will be interested in some information on the design and life of farm tractors developed as a by-product of studies of tractor use and fuel economy by USDA's Bureau of Agricultural Economics.

This study shows that of the wheel tractors made in 1938 about 96 percent were still working after 10 years. Of the machines sold five years earlier (1933) only about 15 percent had gone out of service. The Bureau reports that on the basis of its findings, farm tractors have a life of 16 to 22 years.

The average life of tractors has increased about 50 percent since 1941, USDA reports. The life of current models may be even longer. Rubber tires on current tractors reduce vibration and this tends to prolong years of service.

Gas Turbine Use Seen For Bi-Metals

FARMINGDALE, N. Y.—Bi-metallic gas turbine stator blades, having aluminum alloy roots for easy machining, are under study in England according to an announcement of Wellworthy Pistons, Ltd., holder of the British license for the molecular bonding of aluminum to steel by means of the "Al-Fin" process. Primary object of the study is the development of a rapid, economical method of manufacturing compressor stator assemblies. Wellworthy is licensed by the Al-Fin Division of the Fairchild Engine and Airplane Corp., developer of the patented molecular bonding process.

Bright drawn steel strip is used for the airfoil section. The strip,

which can be obtained to extremely close dimensions from the mills, is cropped to length and—by means of the Al-Fin process—an aluminum alloy root is bonded to it. The alloy used is selected for its corrosion resistance and strength. The immediate advantage of the method is that it provides a means of manufacturing compressor stator blades with a root section that is readily machineable—hence less expensive. In addition, development is foreseen of compressor stator blade units in segments containing a number of blades or even complete stage rings with blade sections made integral by bonding in of the drawn airfoil strip steel. Cost and weight advantages are foreseen as resulting from such construction. Extensive testing has proved the strength of the Al-Fin aluminum-to-steel bond. The Al-Fin bond has a tensile strength of approximately 15,000 psi. Tensile tests at elevated temperatures show that the bond retains its strength at temperatures at least as high as the 600 to 700 degrees F encountered in compressors. The bond is vibration proof and is leakproof to hot oil and diatomic gases.

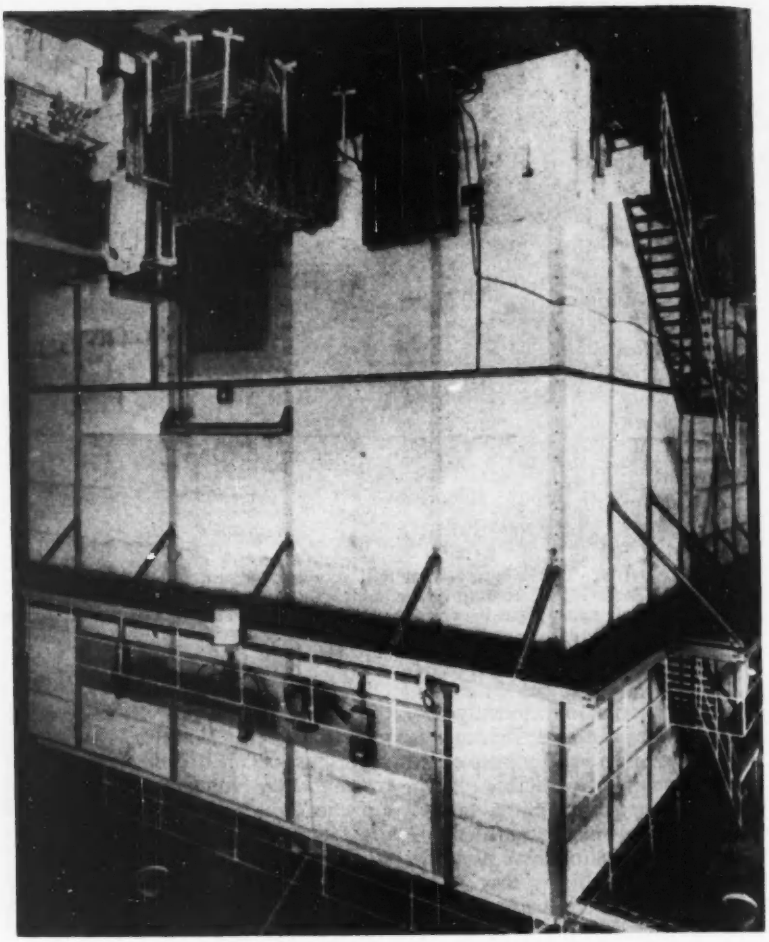
Pure aluminum and any of the common casting alloys such as 43, 142, 195, 355, 356, Almag 35, Frontier 40E, etc., which have much higher tensile strength than pure aluminum, can be bonded to cast iron, "Ni-resist", carbon, alloy, stainless, inconel and the nimonic series. It is possible to heat treat after bonding, in some cases.

Besides applications such as that being investigated by Wellworthy, the Al-Fin process is being used extensively for such things as aluminum-finned steel cylinder barrels, steel-backed aluminum bearings, bi-metallic pistons with ferrous ring carriers, aluminum gears with bonded-in-steel hubs, lightweight aluminum housings with steel or iron bearing retainers, and bi-metallic brake drums.

National Science Foundation Gets $225,000 For First Year

WASHINGTON—When President Truman signs the $17,000,000 supplemental appropriations bill, $225,000 will become available for the first year's operation of the new National Science Foundation.

One of its jobs will be to keep a roster of the nation's scientific, technical and engineering personnel. This roster will be invaluable in case of an all-out mobilization and might be used to draft technical men if war comes.



Atom to Electricity? Not Yet, Say AEC

CHICAGO—An over-enthusiastic newspaper report to the effect that atomic energy was now being converted directly to electric power had technical men on the edges of their chairs, early this month, until an emphatic denial from the atomic energy commission broke the bubble.

The device referred to in the original report, a boron-coated thermopile, or neutron thermometer as it more often is called, is built on the well-known principle of the thermocouple for measuring minute variations in temperature. It has not been and cannot be used for direct or indirect conversion of atomic energy into electricity in anything approaching sufficient quantities for power purposes, according to AEC. Nor has it been used in any plutonium plants to increase plutonium production, as was stated in the Chicago report.

Looking at it practically, AEC spokesman explained that a neutron thermometer could generate less than enough electricity to light an ordinary flash light bulb; it would take approximately 4,000 thermometers to generate one kilowatt of electricity. Furthermore, the neutron thermometer is made up of fine wires, some of them boron-coated. Boron absorbs neutrons that would be necessary for chain reaction, which in itself would stop production of atomic energy. It is foolish to claim, as did the Chicago story, that this thermometer "can help reduce atomic furnaces to one fifth their present size."

Spokesman for the commission stated that there is no practical method known at present to convert atomic energy to power uses. The only possibility seems to be in developing a method for operating an atomic furnace at a high temperature and using the tremendous heat generated to drive a turbine.

Equally absurd was the original report that the instrument will be "one of the key factors on which AEC scientists are said to be depending in their previously announced efforts to build an atomic furnace small enough to power a submarine."

Stoves For Combat Planes

DAYTON, OHIO—Improved electric ovens for giant bombers and troop carriers on long flights have been developed here at the Wright-Patterson Air Force Base. They will provide the means of supplying hot food to the largest crews even during a round-the-world non-stop trip.

Along with the development of these ovens, new types of canned rations have been developed. Precooked canned meals, known as IF-2 rations, contain such foods as boned chicken, beef and pork loaf, ground meat and spaghetti. They contain also fruit, crackers, cookies and candy for dessert. The frozen dinners contain meat, potatoes and a green vegetable, all wrapped in aluminum trays.

National Award Program Fosters Design Improvements

INSTITUTION OF A NATIONAL AWARDS PROGRAM, with $1,000 in cash prizes offered for technical papers dealing with new designs, redesigned steel castings, conversions and advanced uses for steel castings, including improved materials and methods applications to meet specific engineering problems has been announced by Steel Founders' Society of America.

With headquarters in Cleveland, the Society's membership comprises a majority of the steel foundries in North America and more than 90 percent of actual productive capacity. Its major activities include fostering intensive research and product development programs intended to provide practical solution to various design and manufacturing problems; developing better understanding of the steel casting processes through dissemination of technical information; and effecting material improvements, through use of steel castings, in serviceability, appearance and basic costs of parts now produced by other methods.

Basic aim of the national award program, closely related to current advances being achieved through the Society's intensive research and product development undertakings, is to stimulate unbiased and original thinking and practical applications in regard to parts or products which can be made advantageously as steel castings. Particular emphasis is directed to development of new applications or uses for steel castings, including the selection of material and method which will best meet engineering, manufacturing, and sales requirements for a specific part or product.

In all, 15 awards totalling $1,000 will be made, with a top award of $300 for the paper adjudged most useful, two additional awards of $200 and $100 each, four others in amount of $50 each, and eight $25 awards. The program is open to all persons except employees of Steel Founders' Society of America and of companies which are members of the Society. Closing date for the competition is December 1, 1950.

Chemical and Plastics Associations Consolidate

WASHINGTON — The Manufacturing Chemists' Association, Inc. and the Plastic Materials Manufacturing Association, Inc. in regular meetings last week voted unanimously to consolidate. The new association under the name of the Manufacturing Chemists' Association, Inc. has now been organized under the laws of the State of New York, all members of the former associations becoming members of the new corporation.

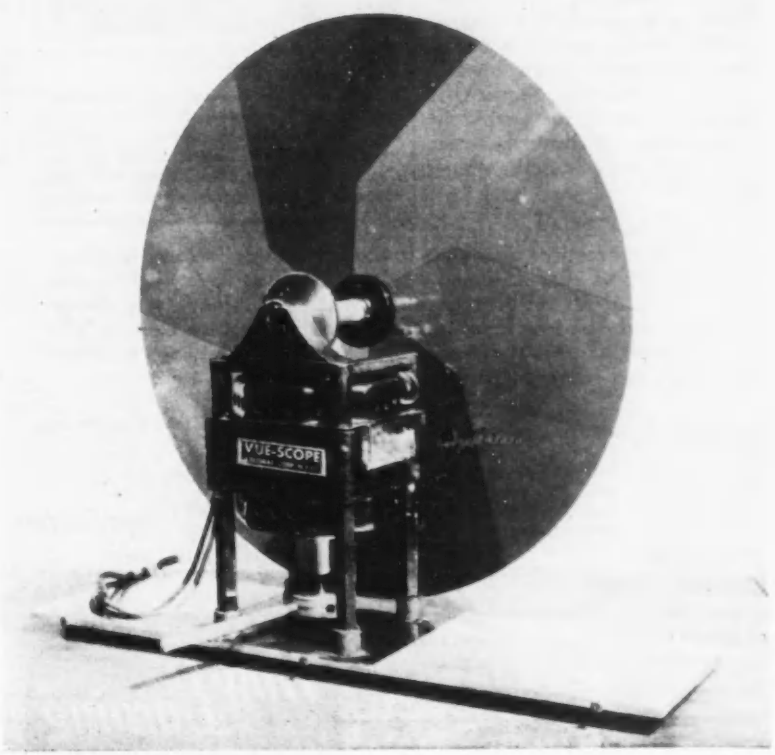
All the officers of MCA were re-elected for the year 1950-1951. The consolidated associations will continue to coordinate activities of plastic materials manufacturers as done by PMMA in the past, and in over-all industry problems they will operate as a group within MCA; the former directors of PMMA will act as a steering committee for plastic activities.

Commerce Department Running Technical Information Service

COMMERCE DEPARTMENT'S OFFICE OF TECHNICAL SERVICES is inviting inquiries for its technical reference service. The object is to generate more business for its collection of 200,000 technical reports. OTS sells the reports in printed and mimeographed forms.

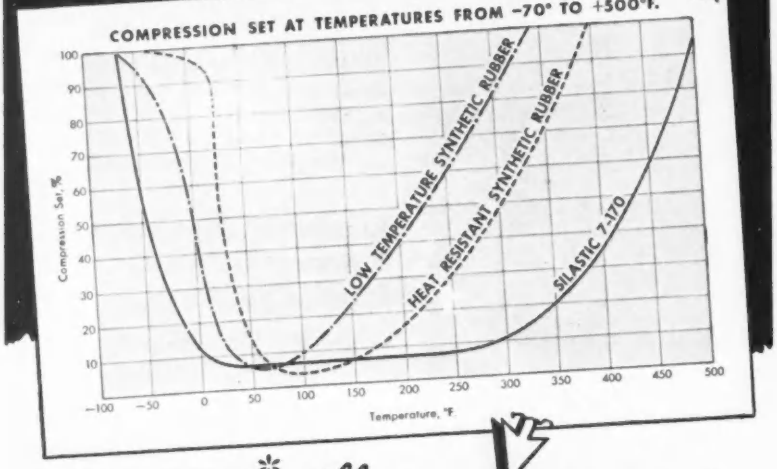
"If you can't come to Washington," OTS advises businessmen, "send your inquiry by mail.

"Technical librarians will check the five hundred subject catalog drawers from 'A Alloy to ZZ model engine'—and on many subjects, mimeographed bibliographies are kept in stock."



COLOR DISK of the Monsanto Fibestos cellulose acetate plastic sheeting made up of three primary colors, red, blue and green, is included in converter kit to be used with sets adapted for CBS color TV to give a 6 in. picture. The kit will introduce TV owners to color until permanent 12 in. converters are offered.

Where ordinary rubber gaskets fail . . .



SILASTIC* still stays Elastic!

AT EXTREME TEMPERATURES, Silastic has greater resistance to compression set—or to permanent deformation due to heat and pressure—than any other rubberlike material. Its elastic memory exceeds that of both the best low temperature and the best high temperature organic rubbers available. Silastic 7-170 forms a more resilient seal at −50°F. than a special low temperature organic rubber does at −7°F. At 450°F., Silastic has more resistance to permanent compression set than the most heat-stable organic rubbers have at 330°F.

PHOTO COURTESY CONSOLIDATED VULTEE AIRCRAFT CORP.

In aircraft cabin heating and pressurizing systems, Silastic gaskets stay elastic under operating temperatures ranging from −70° to 400°F. Similarly, Silastic gaskets and O-rings withstand hot oils at about 450°F. in automotive, aircraft, diesel-electric engines.

Combine that kind of elastic memory with excellent resistance to aging, to oxidation and to attack by a variety of chemicals and hot oils, and you have Silastic—the most stable of all resilient gasketing materials. That's why design engineers and maintenance men specify Silastic, the Dow Corning Silicone rubber that pays for itself many times over in reduced maintenance costs and improved performance.

SEND TODAY!
For your copy of Silastic Facts No. 10 containing new data on the properties, performances and applications for all Silastic stocks.

*T. M. REG. U. S. PAT. OFF.

from +500°F. SILASTIC stays Elastic to −100°F.

DOW CORNING CORPORATION, DEPT. R-23, MIDLAND, MICH.
Please send me Silastic Facts No. 10
Name_____
Company_____
Address_____
City_____ Zone ____ State_____

Dow Corning
FIRST IN SILICONES

Atlanta • Chicago • Cleveland • Dallas • Los Angeles • New York • Washington, D. C.
In Canada: Fiberglas Canada Ltd., Toronto • In England: Albright and Wilson Ltd., London

SILICONE NEWS

Dow Corning Silicone Insulation Saves Over $8,000 a Year on 4 Motors!

In the Saginaw Plant of Eaton Manufacturing Co., the output of four of these machines is stepped-up by stopping the 5 h. p., 3600 r.p.m. motors with a jolt of direct current as soon as 5 operations are performed on a small automotive part. With 105 starts and stops per hour, Class A insulated motors had an average life of only 8 days—32 failures for each of the 4 motors or a potential $10,000 a year in rewind costs alone.

Rewound with Dow Corning Silicone (Class H) Insulation, these motors now have a minimum life of 6 months; save Eaton a potential $8,000 a year plus a few hundred man hours of labor and some 40,000 parts that would have been lost due to down time.

DOWN TIME AND LOST PRODUCTION DUE TO MOTOR FAILURE ARE NO LONGER NECESSARY

You can increase the life and reliability of almost any product that is electrically operated or controlled by specifying Class H insulation made with Dow Corning Silicones. For data on the properties and performance of silicone insulation write to Department R-23A, Dow Corning Corporation, Midland, Michigan.

REMEMBER
Dow Corning Silicones Mean Business

Properties of Rectangular Sections

C. A. KULMANN

A SINGLE LINE DRAWN between the height and width of a rectangle on this chart gives: area, section modulus and the moment of inertia. Inertias can be obtained when either the axis passes through the center of gravity or when the axis coincides with one edge. In calculating the moment of inertia of a built-up section, this nomograph can be useful by dividing the cross-section into rectangular elements and using the Parallel Axis Theorem:

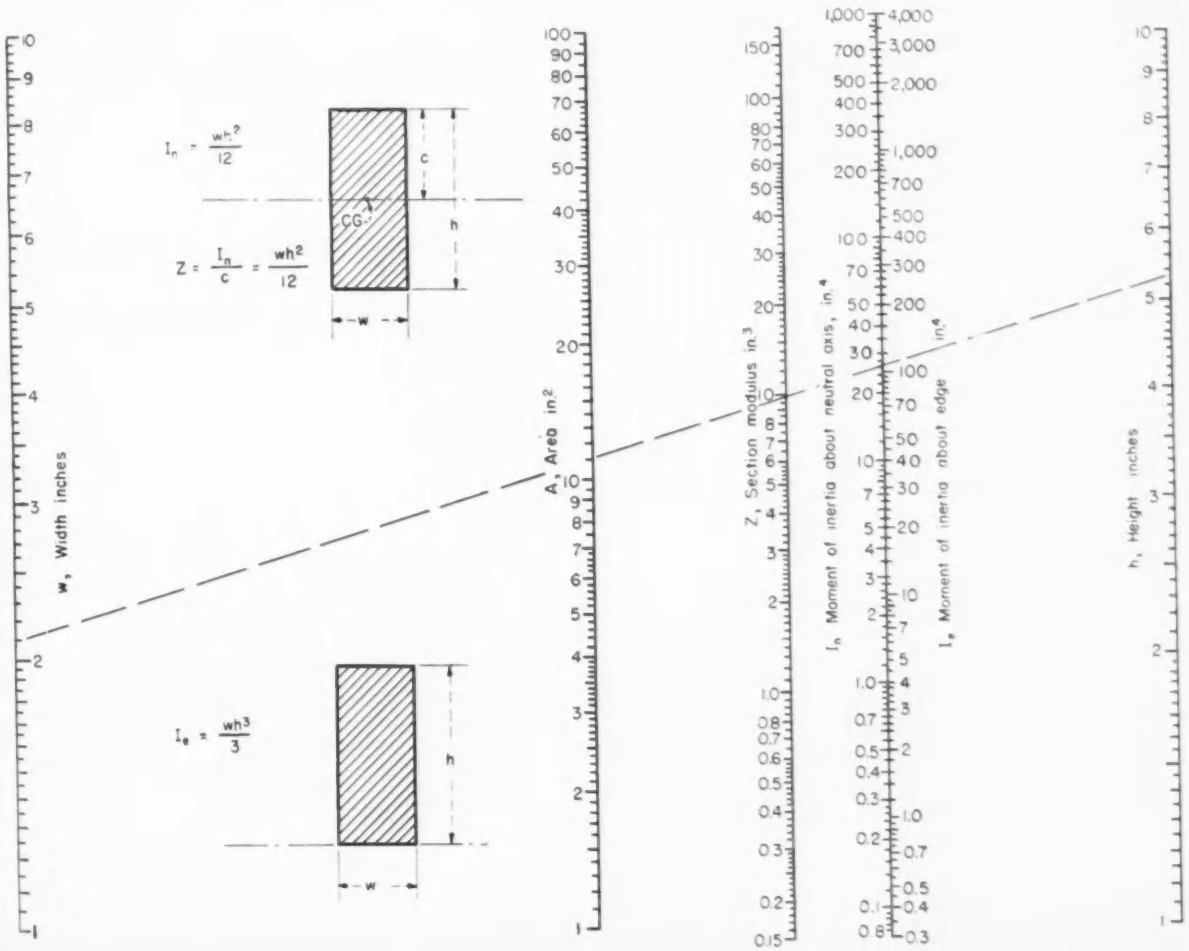
$$I_o = I_n + Ad$$

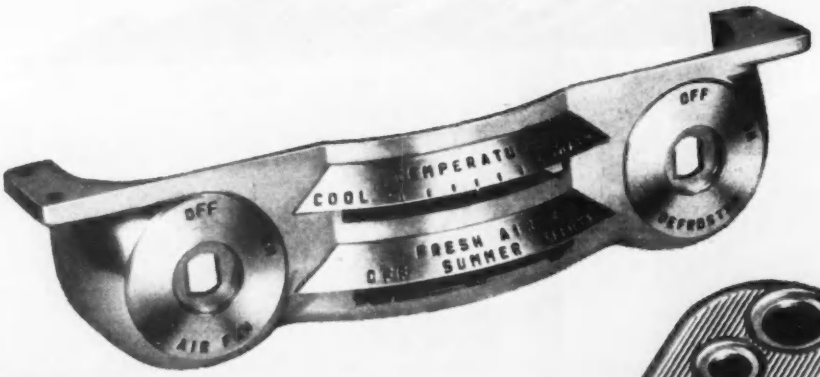
where, I_o is the moment of inertia of the elemental section about the distant axis which is sought; I_n is the moment of inertia about its c.g (from the chart); A is the area of the element; and d is the distance between the parallel axes.

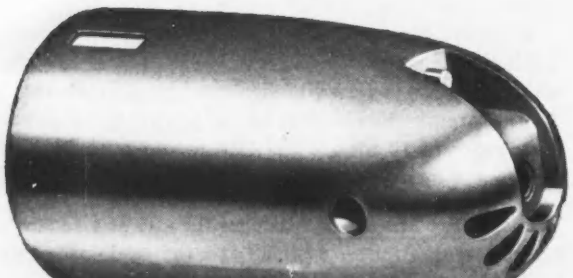
Whenever the dimensions of the rectangle are beyond the limits of scale, use the scale with a modified value which has been increased or decreased by a factor of 10. The final results should then be correspondingly corrected by: a factor of 10 for the area; 10^2 for the modulus; and 10^3 for the inertias.

Example: The line drawn illustrates the case of a rectangle 5.3 by 2.1 in. Area = 11.1 in.^{2}; section modulus = 9.8 in.^{3}; Moment of Inertia, I_n = 26.0 or I_e = 104 in.^{4}.

This chart is one of many included in the book "Nomographic Charts" which will be soon published by the McGraw-Hill Book Company.







AUTO-LITE

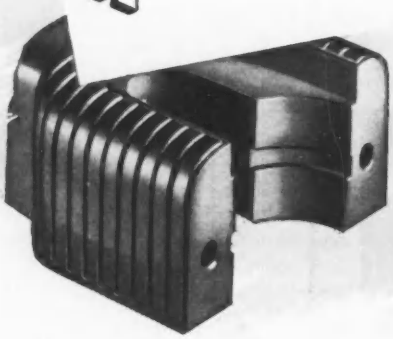
DIE CASTINGS

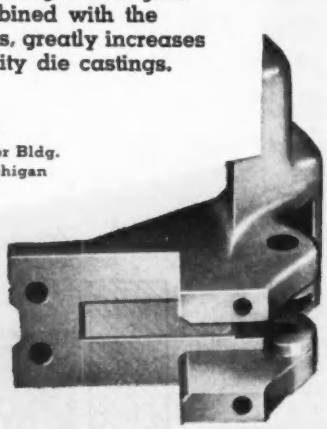
● Increased production facilities for die castings made possible through the famous Auto-Lite "controlled metals" processes are now available. The opening of the great new Lockland plant of Auto-Lite, combined with the enlarged facilities at Woodstock, Illinois, greatly increases Auto-Lite's ability to furnish high quality die castings.

THE ELECTRIC AUTO-LITE COMPANY
Die Casting Division, Woodstock, Illinois

600 S. Michigan Ave.
Chicago 5, Illinois

723 New Center Bldg.
Detroit 2, Michigan





Inquiries invited

Catalog will be sent on request.
Write on your company letterhead.

Conversion Chart for Dimensional Tolerances Angular and Linear

KENNETH F. STONE
Chief Production Engineer
Dictaphone Corporation

IN DIMENSIONING MACHINE PARTS for production, a conversion is often required to change from linear to an angular tolerance. Such a change can be easily made by use of this chart. For greater accuracy this graph is divided into two scales: radial distances of less than 1 in. and a range from 1 to 10 inches.

EXAMPLES: A rocker cam is illustrated in Fig. 1 where important dimensions have been stipulated as ±0.002 in. linear tolerance. To locate the holes shown, two dimensions are required: the distance between the holes, and the angle. With the radial distance of 2 in. (between holes), the chart indicates that an angular tolerance of ±0°—04′ is required.

To properly specify that the straight cam surface is to be parallel within ±0.002 in., an angular tolerance is required. Since this surface is about 1 in. long, the permissible angular variation is given on the chart as ±0°—07′.

This graph can be also helpful in converting angular to linear tolerances. Fig. 2 shows a milled slot that is dimensioned with an angular tolerance of ±2 deg. Since the radial distance is approximately 0.10 in., this tolerance is equivalent to a linear variation of ± 0.0035 at the outside surface.

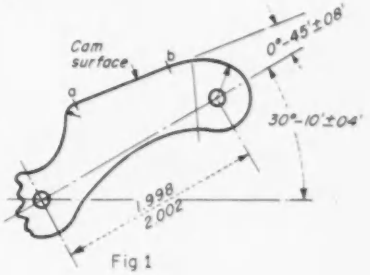


Fig 1

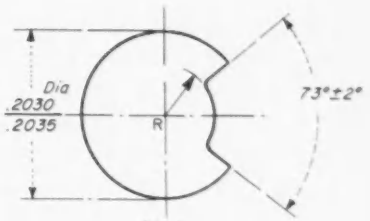
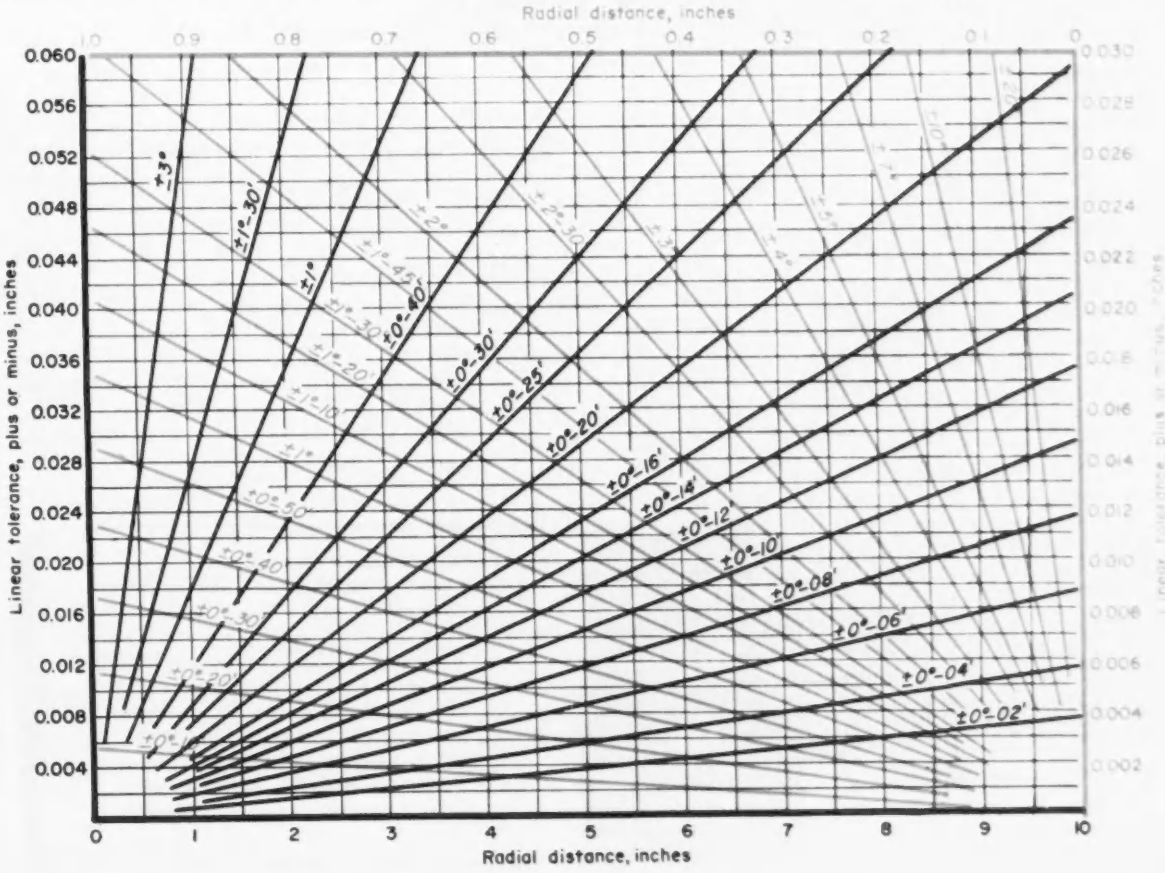
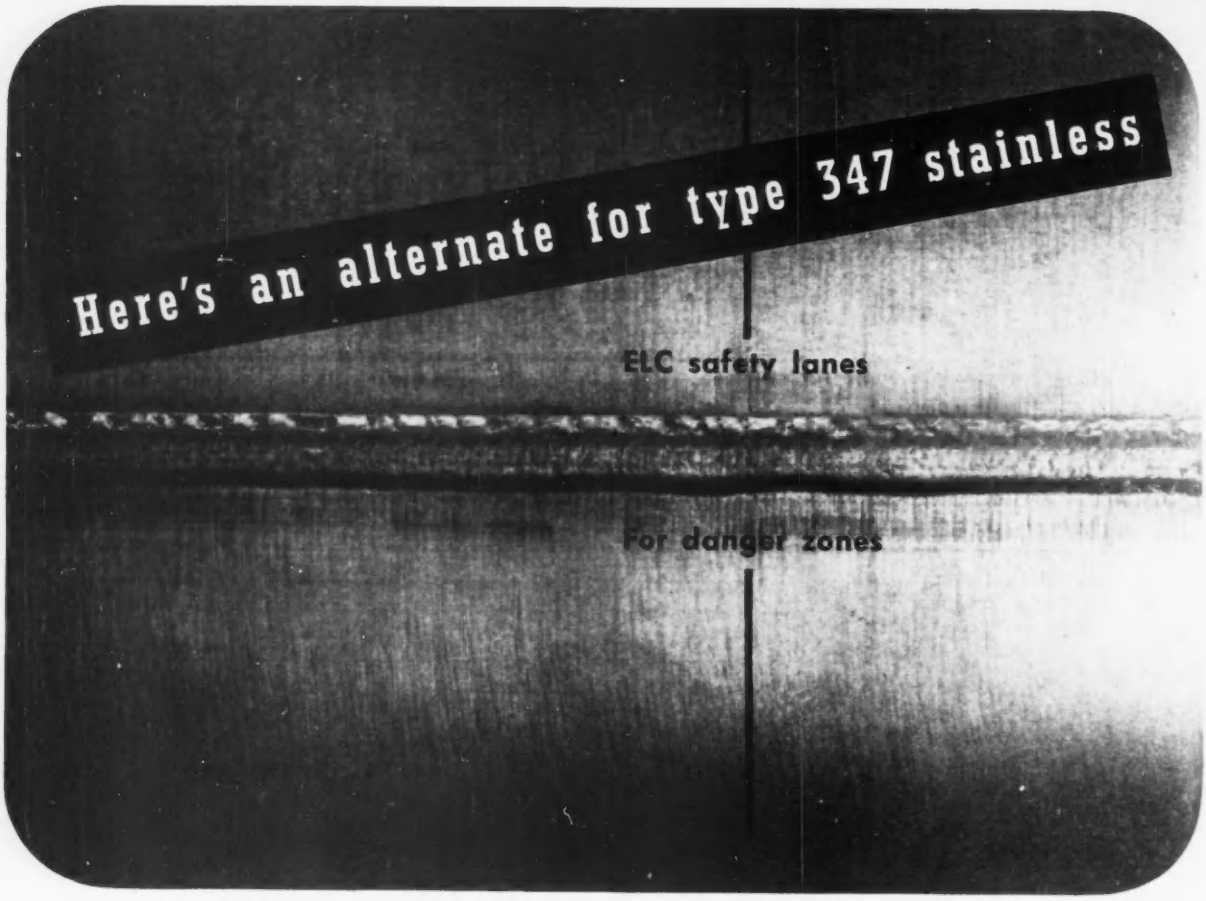


Fig 2





If the shortage of columbium is putting Type 347 Stainless Steel out of your reach for important defense work, here's a special Armco grade that may be a suitable alternate. It is Armco Type 304 ELC Stainless Steel.

This Extra-Low Carbon grade can be used in place of columbium-stabilized stainless for welded equipment exposed to service temperatures below 800°F. Above this temperature, Type 321 should be used.

Stabilization with columbium or titanium is not needed with Armco Type 304 ELC because of its low carbon content (.03% maximum). This eliminates intergranular corrosion adjacent to the welds, because no harmful carbide precipitation occurs during welding. Annealing after welding is not necessary, except where engineering requirements specify stress-relief after fabrication.

Armco ELC Stainless sheets, strip and plates have proved satisfactory for many applications in the aircraft, chemical, oil, textile, paper, and allied industries.

other types

Extra-Low Carbon Stainless is also available in Armco 18-12 Mo ELC (Type 316 ELC) and Armco 19-12 Mo (Type 317 ELC). Mechanical properties of the ELC grades are similar to those of the corresponding standard grades.

Armco engineers will be glad to talk with you about specific applications for the ELC grades. Just fill in and mail the coupon.

ARMCO STEEL CORPORATION

6080 Curtis Street, Middletown, Ohio
Plants and Sales Offices from Coast to Coast • The Armco International Corporation, World-Wide.

ARMCO

Send me Data On "ELC"

ARMCO STEEL CORPORATION
6080 CURTIS STREET, MIDDLETOWN, OHIO

We manufacture_____

Name_____ Title _____

Firm Name_____

Street Address_____

City_____ Zone___ State_____

Column Nomograph

C. A. KULMANN

THIS CHART should be useful in solving problems involving simple columns, struts or links; where standard or heavy pipes and square or solid shafts are used. It is based on the Rankine formula:

$$s = \frac{P}{A}\left[1 + K\left(\frac{l}{r}\right)^2\right]$$

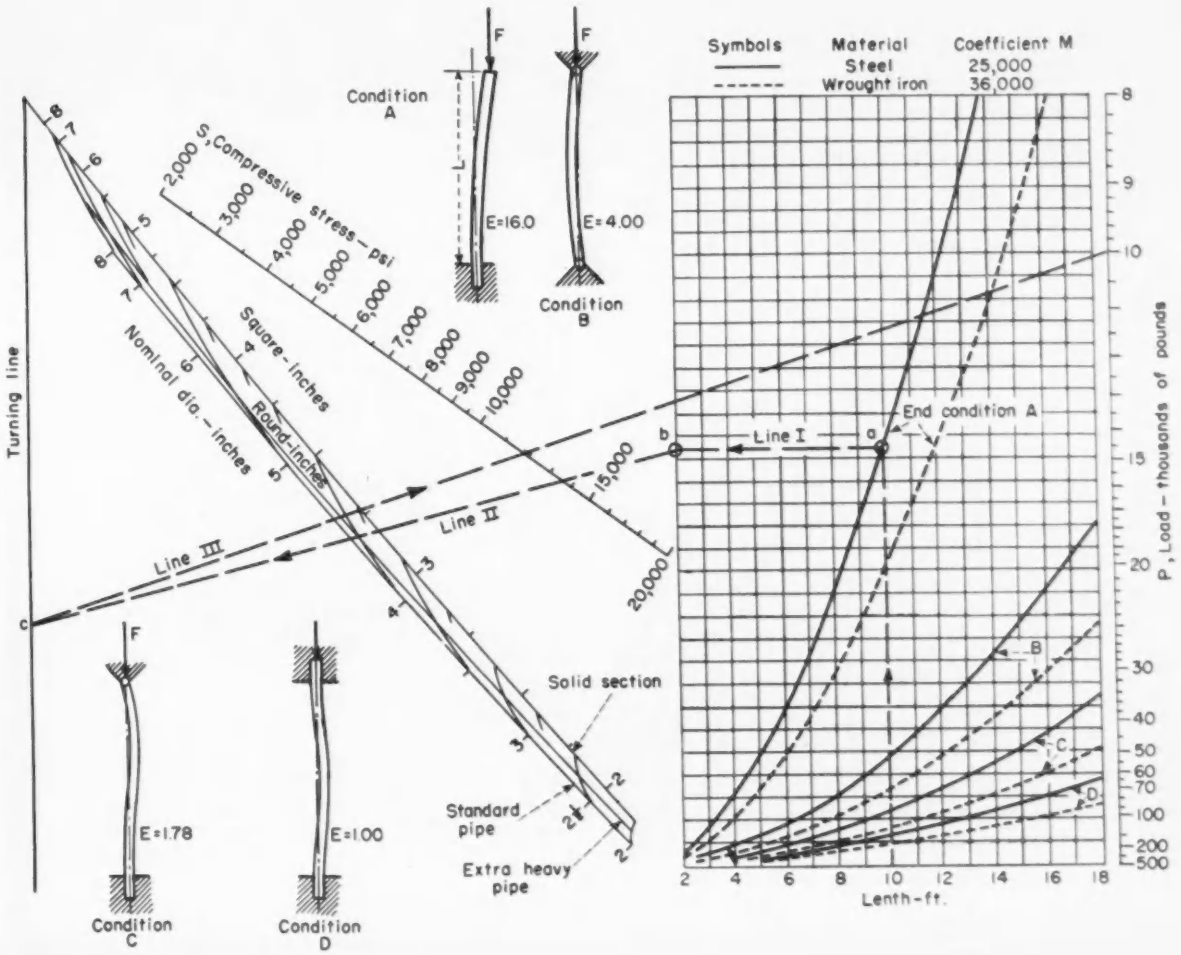
and applies to columns with a slenderness ratio (l/r) of from 20 to 120. In this equation, s is the allowable compressive stress, psi; P is the load, lbs; A is the cross-section area, sq in.; K is a factor which is based on the material and end conditions of the column; l is the length, in., and r is the radius of gyration of the cross section, inches.

This chart is divided into two sections: a graph and a nomogram. The graph accounts for the material, length of column and end-conditions. Included is the factor, $K = E/M$, where E is the end condition coefficient and M is the material coefficient. The nomogram has a scale which defines the cross section of the column; note that the upper line applies to solid bars. On this line, the outer scale corresponds to the side dimension of a square section while the inside scale applies to the diameter of the round members.

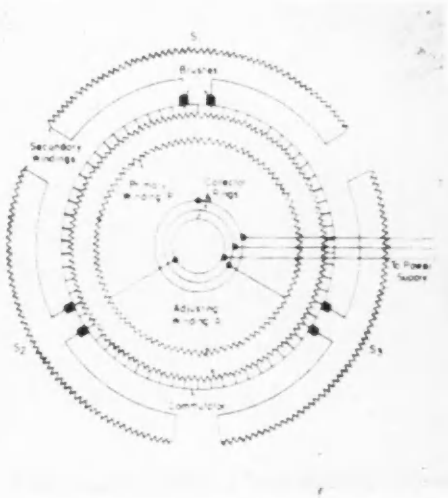
EXAMPLE: Find the compressive stress in a 10 ft. column made of a 4 in. extra heavy steel pipe with one end fixed and the other free under a load of 10,000 lb.

The length, end-conditions and material determine point *a*. From this point, Line I is projected horizontally toward the left to point *b*. From point *b*, Line II is drawn through the cross-section scale (4 in. extra heavy pipe) to the Turning Line, point *c*. From this point, Line III is drawn to the load scale (10,000 lb). This line intersects the compressive stress scale at 11,800 psi.

This graph is one of many included in the book Nomographic Charts, which will soon be published by The McGraw-Hill Book Company.



NEW COMPONENTS-



A PILOT MOTOR is actuated by the mechanical follow-up control. On successive starts it automatically drives the motor brushes to a setting corresponding to the speed set on the adjustment knob. Stop or automatic slowdown returns the brushes to the lowest speed position without disturbing running speed setting. There are no secondary resistors to waste energy.

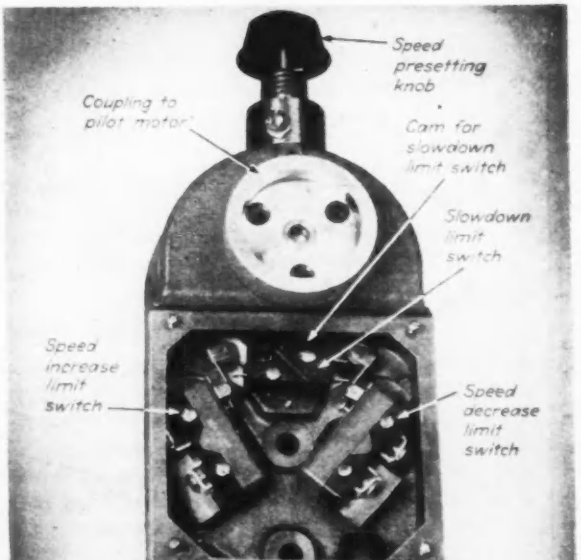
Mechanical Device Fixes Motor At Present Speed

An a-c adjustable-speed motor is available with a mechanical follow-up control for achieving a fixed preset speed without the use of expensive control equipment. The device enables a predetermined speed to be set manually by a knob either directly on the motor or at the end of a flexible cable 10 to 15 ft long. The adjustment can be made while the motor is at a standstill or while it is running.

Speed control is obtained by rotating brushes around the commutator. Since speed is a function of brush position, any device that will automatically return the brushes to an adjustable, predetermined position on successive starts will serve the desired purpose of establishing preset speed operation.

In operation, the mechanism actuates a pilot motor which drives the brushes to a position corresponding to the setting of the speed adjustment knob. A stop or automatic slow-down returns the brushes to the lowest speed position without disturbing the original setting. When the motor is restarted, the brushes again come back to the preselected speed without any attention from the operator. Accelerating time is dependent upon pilot motor speed.

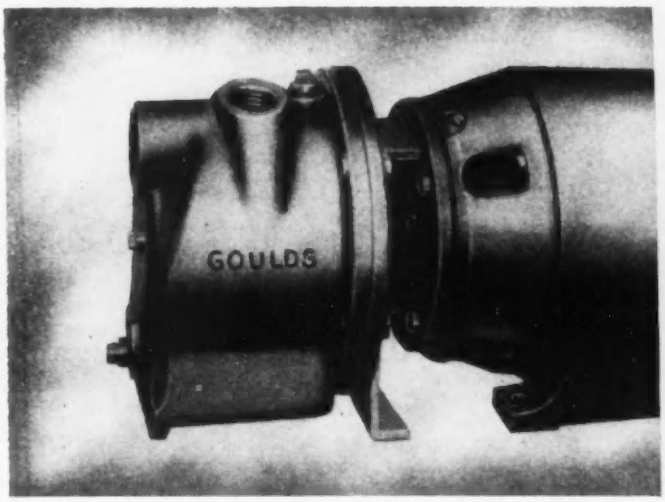
Some of the applications for the motor with the speed-setting device are on offset printing presses, can-sealing machines, planer-feed-roll drives, conveyor drives, ball mills, small rolling mills, mixers, punch presses and fan drives. Motors are available in a variety of power ratings.

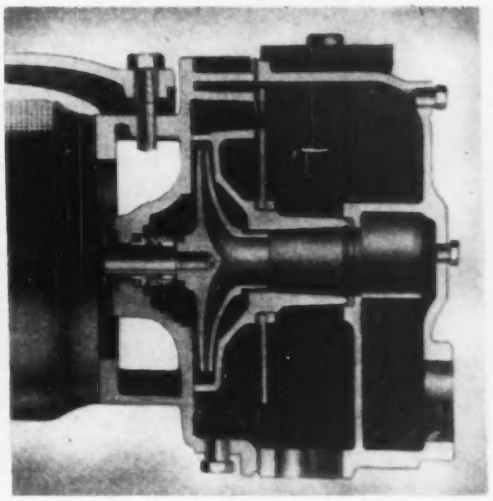


BRUSHES ROTATE around the commulator to control speed. A small control motor drives the brushes through gear head. The control knob moves the running-speed stops to set final motor speed. When motor stops brushes are returned by control motor to starting position but knob setting is uneffected. Running speed on restarting is not changed. Assembly view above shows switch box with flexible shaft in place.

General Electric Co., Schenectady 5, N. Y.

Circle No. 1 on Reader Service Card





FAST PRIMING PUMPS have efficiencies comparable to standard centrifugal pumps. They will not vapor lock and no suction check valves are necessary. Capacities are ½ to 5 hp at 3,500, and in some cases, 1,750 rpm. Volatile liquids, oils and water are handled. Models without motors ride on two sealed ball bearings. Bellows-type mechanical seal adjusts for wear.

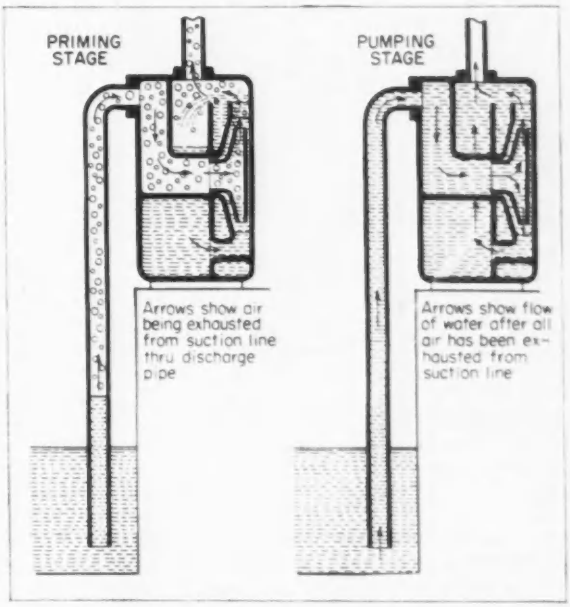
Centrifugal Pump Primes Itself

The fast, positive self-priming action is due to the design of the volute or guide vane. This volute design enables the pumps to exhaust a suction line of air with only an initial priming charge of the liquid in the casing.

The guide vane has two volutes with cut-waters unequally spaced in respect to circumference of the impeller. The cut-water of the upper volute is in very close proximity to the impeller vane tips while the cut-water of the lower volute is at a much greater distance from the impeller. The purpose is to create an unbalanced condition so that after the pump casing is filled with liquid and the pump started, and during the suction line priming phase of its operating cycle, liquid will flow into the lower volute past the wide cut-water, while liquid and air will flow out of the upper volute at the close cut-water.

The upper volute discharges liquid and entrained air at atmospheric pressure into the air separation chamber where its velocity is retarded, permitting the air to separate from the liquid and pass out through the pump discharge. The air-free liquid is recirculated into the other volute. No recirculation of the liquid through the impeller eye is employed. The liquid cannot enter the impeller eye except through the suction line.

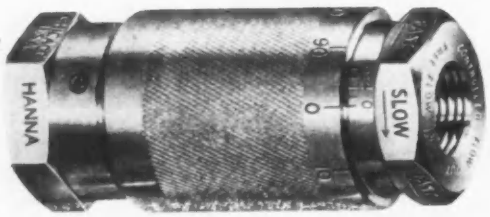
This operation continues until all air has been exhausted from the suction piping. When the liquid rises and enters the pump section, it flows directly into the eye of the impeller as in a conventional centrifugal pump.


PRIMING STAGE
PUMPING STAGE
Arrows show air being exhausted from suction line thru discharge pipe
Arrows show flow of water after all air has been exhausted from suction line

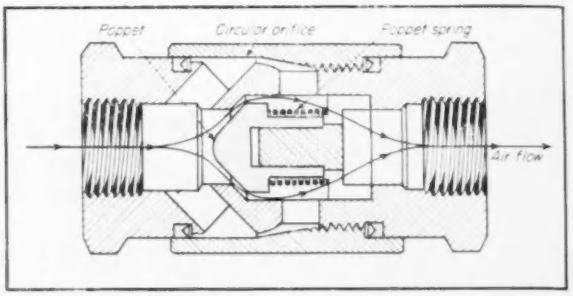
LIQUID ENTERS IMPELLER eye only from suction line. Air rises to top and is pushed out discharge line. Liquid trapped in the bottom of the casing prevents recirculation of the air. No valves are required to keep priming liquid in casing. The cut-water is closer to the impeller blade tips in the upper guide vanes than it is in the lower. In priming, liquid will flow in the lower portion past the wide cut-water, while entrapped air will pass out the top at the close cut-water. Integral motor or separate pumps are offered for 5 to 120 gpm cold water ratings. *Goulds Pumps Inc., Seneca Falls, N. Y.*

Circle No. 2 on Reader Service Card

CONTINUED ON PAGE 174



MICROMETER GRADUATIONS for knurled hand grip allow close control of actuating cylinder speed. Valves can be used for air, water or oil.



POPPET OPEN, as shown, allows flow through opening equal to full pipe area. Exhaust flow, with poppet closed, is through adjustable circular orifice.

Adjustable Valve Controls Exhaust Speed

A line of speed control valves is available in three sizes, $\frac{1}{4}$, $\frac{3}{8}$ and $\frac{1}{2}$ in., with micrometer graduations for precision speed control of low pressure cylinders. A fourth $\frac{1}{4}$ in. valve is specifically designed for use with tubing and functions in the same way as the larger valves but does not include the micrometer adjustments.

The valves are suitable for air, water or oil control, with maximum pressure to 250 psi, and maximum temperature to 250 F. They may be mounted in any position.

The face of one end is marked "Controlled flow out—Free flow in" and the other face is marked "Controlled flow in—Free flow out." Controlled flow means that the orifice controlling the flow can be adjusted from zero to full size opening—with one revolution of the adjustment sleeve. Graduations are in hundredths of a revolution, with each hundredth representing 1/100 of the pipe area. Free flow means unrestricted flow through an orifice equal to the pipe area.

Valves are designed so inside construction is not damaged if excessive torque is applied during installation. All metal parts are made of aluminum except for stainless steel springs. Seals and poppets are of molded synthetic.

Hanna Engineering Wks., 1765 Elston Ave., Chicago 22, Ill.

Circle No. 3 on Reader Service Card

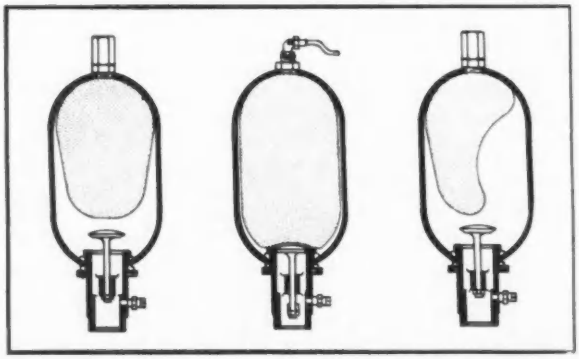
Poppet Valve Prevents Damage to Bladder

The accumulator stores hydraulic oil under pressure for the actuation of hydraulic devices where the demands of the system are intermittent. It provides high oil flow capacity (hydraulic horsepower) for relatively short cycles of operation.

Volumes within the range of hydraulic pressure drop are instantaneously available providing there is adequate time between cycles for the pump to recharge the accumulator. The accumulator is also used to help maintain uniform pressure over extended work cycles. It will serve as a shock absorber and to compensate for variations in systems having thermal expansion. It will also compensate for leakage in closed circuits.

An accumulator with suitable unloading valving makes possible the storage of oil under pressure with minimum operation of the hydraulic pump under pressure. Resulting power input savings are appreciable when system demands are infrequent and the volume requirements for the operation of the hydraulic devices are such that several operations require only one charging of the accumulator by the hydraulic pump.

The one-piece and seamless alloy steel shell is designed for a high hydrostatic bursting pressure; all sizes are tested to 8,500 psi. Because of the swedged or rolled neck con-



struction, pressure in excess of 15,000 psi causes the lobe (or dome of the cylinder where it fits around the neck) to expand, permitting obvious leakage and retarding further increase in pressure. The result is a high safety factor when using 3,000 psi working pressure.

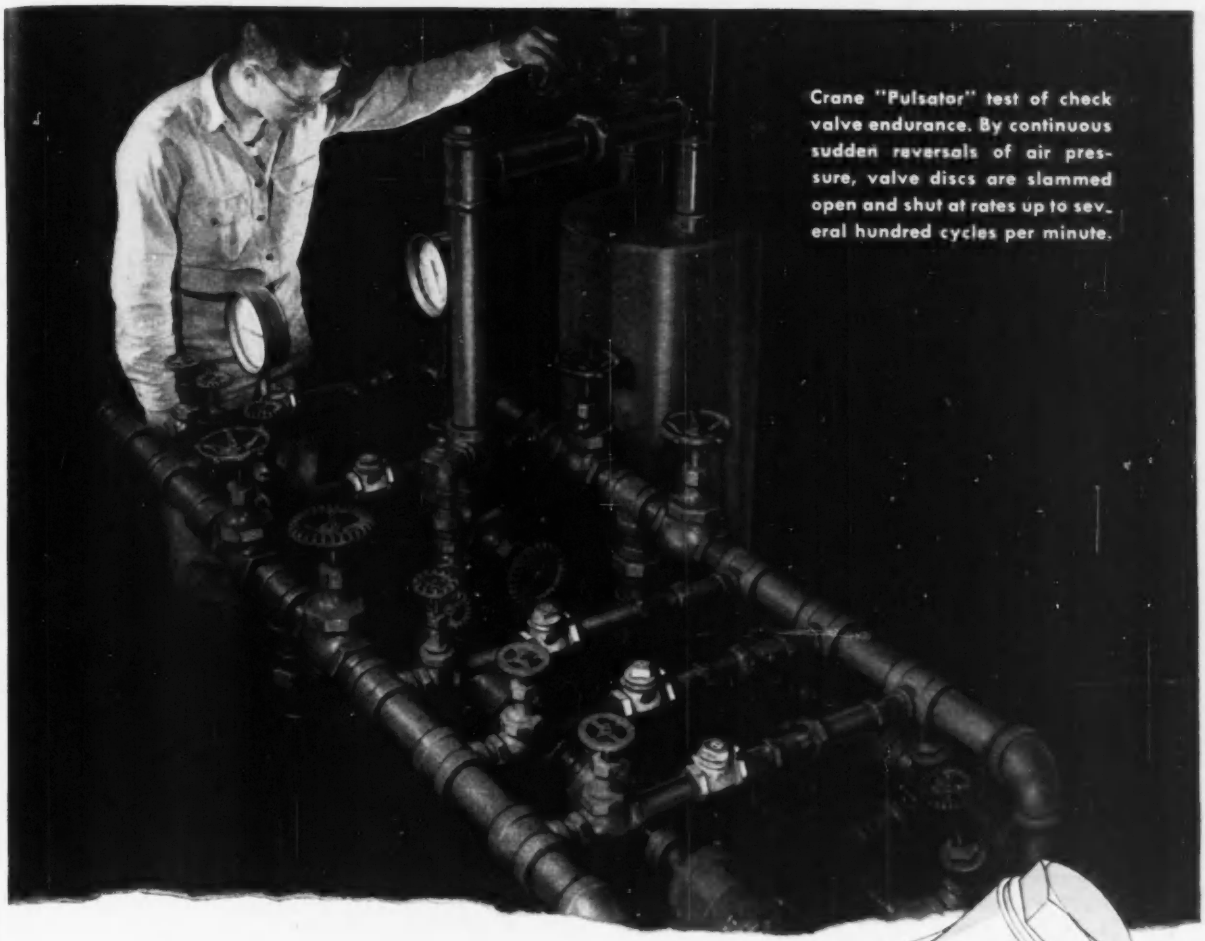
A large-capacity hydraulic connection affords slight restriction to the oil flow discharge. The spring-opening poppet prevents damage to the bladder during discharge.

Vickers Inc., 1400 Oakman Blvd., Detroit 32, Mich.

Circle No. 4 on Reader Service Card

CONTINUED ON PAGE 176

More **CRANE VALVES** are used than any other make



Crane "Pulsator" test of check valve endurance. By continuous sudden reversals of air pressure, valve discs are slammed open and shut at rates up to several hundred cycles per minute.

How Many Checks in a Good Check Valve?

You don't buy them that way, but that's what finally determines the value of check valves in your pipe lines. That's why Crane is so completely equipped for accurately pre-testing valve performance. For instance, "wearability" of check valves is tested by opening and closing the valves continuously on various fluids under conditions of actual service. Special equipment is used to accelerate these tests to equal many years' service in a short period. Countless tests like this—of materials, designs, and finished products—insure the superior performance values that mean lower piping costs for users of all Crane valves and fittings.

This new regrinding Brass Swing Check design typifies Crane's continuing product development for better valve performance, lower piping maintenance costs. For increased flow capacity . . . for fast, positive closure on backflow . . . for easier servicing without removing valve from line . . . get a demonstration of these 200 and 300-Pound Y-Pattern Check Valves. Phone your Crane Representative, or ask for literature.

CRANE VALVES

CRANE CO., *General Offices:* 836 S. Michigan Ave., Chicago 5, Ill.
Branches and Wholesalers Serving All Industrial Areas

VALVES • FITTINGS • PIPE • PLUMBING • HEATING

Vanadium Metal Is Available Commercially

A high-purity, ductile vanadium metal is now available in limited quantities for fabrication and use by the metal industry. The metal averages above 99.8 percent vanadium and can be obtained in the form of ingots, bars, sheet, and foil. It is also available in the form of chips for remelting into ingots of special sizes and shapes.

Physical and chemical properties: pure vanadium has good corrosion resistance, high tensile strength and excellent ductility. It is highly resistant to corrosion by salt spray, sea water and reducing acids in moderate concentrations, and test data indicate it has good anti-fouling characteristics.

Vanadium metal has a modulus of elasticity of about 22 million psi and a density of 6.1. The modulus-density ratio for vanadium is practically the same as that for steel (3.7 million), although the density of vanadium is appreciably lower than that of steel. This compares with a modulus-density ratio for titanium variously given as 3.1 to 3.6 million. From this, it can be seen that vanadium metal has a very favorable strength-weight ratio.

The high modulus-weight ratio of vanadium metal suggests its use for structural parts in which flexural rigidity is the important factor. Its high strength, combined with its resistance to salt water, suggests its use for instrument parts subjected to high dynamic stress in marine atmospheres. The metal also has good static tensile properties which should make it suitable as a spring material.

Fabrication Properties: vanadium ingots can be hot-rolled at temperatures between 1,745 and 2,100 deg F. Rolling practice, with respect to passes and reduction, follows that established for austenitic stainless steels. Sheet bar stock does not work-harden appreciably even during cold-working and may be rolled to a sheet having a thickness of 0.020 in. In fact, a reduction of 85 percent can be made without intermediate annealing.

The machinability of vanadium is about the same as that of cold-rolled steel. The metal is free-cutting and can be machined more readily than nickel, stainless steel, and titanium. High cutting speeds can be used to produce good surface finishes.

Vanadium metal has good workability and may be formed, bent, stamped, and pressed in the usual manner. For example, the Belleville-type springs shown in the accompanying illustration were made in a simple lead-lined die. Vanadium may be welded by the Heliarc process using an argon atmosphere for shielding.

THE HIGH-PURITY VANADIUM METAL is available in the form of ingots, plates, bars, sheet, and strip, as well as chips for remelting. A few fabricated shapes are also shown, including Belleville-type springs, a conical-helical spring, and ring and welded tube. Samples average above 99.8 percent pure vanadium

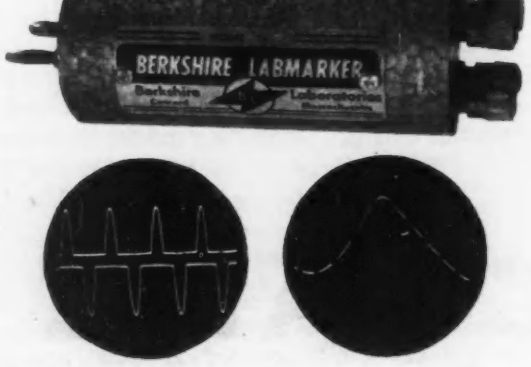
Electro Metallurgical Div., Union Carbide and Carbon Co., 30 W. 42nd St., New York 17, N. Y.

Circle No. 5 on Reader Service Card

Instrument Gives Timing Marks For Oscillograph

A wave-shaping device produces timing marks in cathode ray oscillography. It converts a sinusoidal input of up to 30 volts rms into a series of sharp unidirectional pulses. The unit permits the use of either positive or negative pulses, or both at once. Approximate characteristics are: pulse duration, one-third of a cycle; pulse amplitude, one-half of the rms input voltage; frequency range, 25 cycles to one megacycle. The oscillograms show the output wave forms on a record timed with the instrument. The device was connected to the "Z" axis terminals of the oscilloscope to produce the short breaks in the cathode ray trace.

It is a compact self-contained unit which may be plugged into the terminals of a suitable oscillator. No other power source is required. The output binding posts may be used with leads having single or double banana plugs, spade tips, phone tips, or plain wire ends.



Berkshire Laboratories, P. O. Box 70K, Concord, Mass.

Circle No. 6 on Reader Service Card

CONTINUED ON PAGE 178

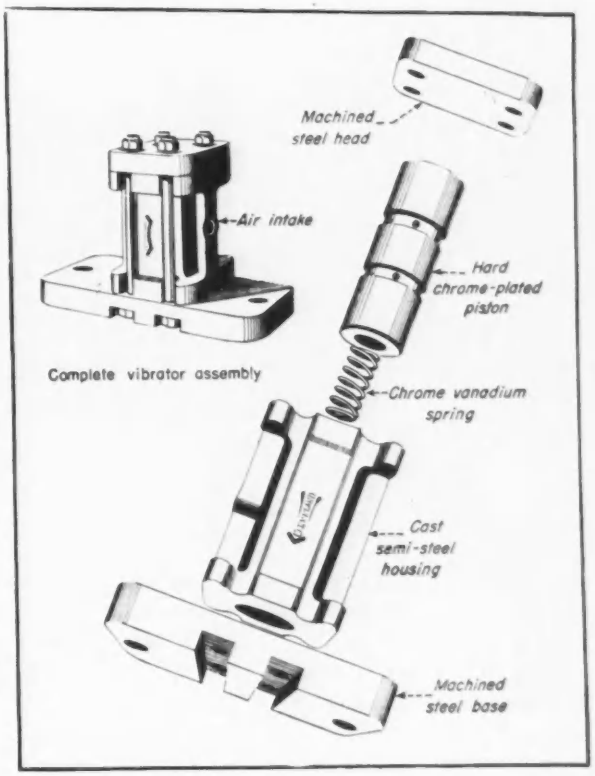
Air Vibrator Shakes Bins and Chutes

The number of major parts in this pneumatic bin and chute vibrator has been reduced to five. Simplified design reduces wear and maintenance and gives easier operation.

The component parts—mounting base plate, housing, spring, piston and head—are shown in sequence diagram. Assembly is kept intact with four longitudinal body bolts and special stop lock nuts. The reciprocating piston is the heart of the vibrator and it develops vibration in the following manner: When compressed air is applied to the intake of the vibrator, the piston is motivated by the air entering the groove exposed to the intake and this air is then passed through the drilled porting in the piston permitting pressure to build up at one end of the assembly, thereby throwing the piston to the opposite end, with the air exhausting en route. Now the other groove in the piston is exposed to the intake and the process is repeated, moving the piston in the opposite direction.

Pistons vary from $1\frac{1}{4}$ to 4 in. in diameter, with $\frac{1}{4}$ to $\frac{3}{4}$ in. intake connection ports. Overall heights range from about 6 to 15 in., with weights from 9 to 185 pounds. Applications are expected in chute and bin systems handling finely pulverized chemicals, powder lime and wet coal to break up arching and ensure steady flow.

Cleveland Vibrator Co., 2774 Clinton Ave., Cleveland 13, Ohio



Circle No. 7 on Reader Service Card

Electronic Tachometer Counts Shaft Revolutions

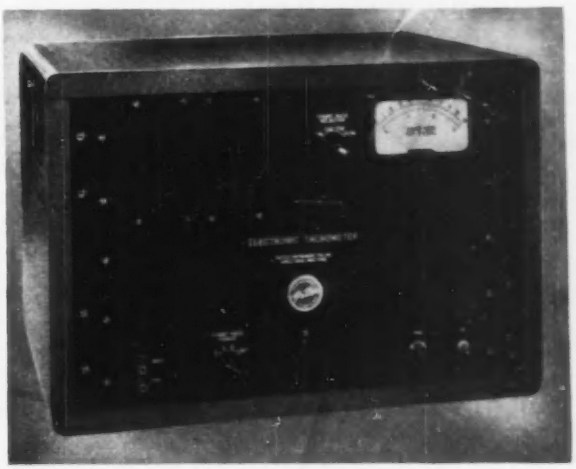
A high speed electronic counter tachometer is designed for high accuracy revolution and frequency measurements. Fundamentally, the instrument totalizes the number of counts derived from the source being measured during a precisely established time interval of 0.6 sec (.01 minute). This reading is displayed on a direct-reading, four-digit electronic counter using ten neon glow lamps for each digit. After each measurement and display period, the instrument automatically resets and recycles. The display time is adjustable over a period of from 0.5 second to 4 seconds, or can be set to hold the count indefinitely.

The 0.6 second time interval is established by counting 60,000 cycles of the 100 KC crystal oscillator contained in the unit. The instrument can be used to measure frequencies up to 100,000 cycles per second with an accuracy of ± 1 cycle. In addition to the electronic counter registration of the measured frequency, the unit also includes a count rate meter which can be used when the unknown frequency source is being adjusted. This type of indication reflects instantaneous changes and is useful when the source of frequency must be manually adjusted to a specific point.

The electronic counter tachometer is suited for high speed turbine, engine, and motor speed measurements and can be supplied with electromagnetic or photoelectric pick-up devices for detecting the speed without physical contact.

Because the instrument is direct reading, it is suitable for laboratory and production frequency measurements.

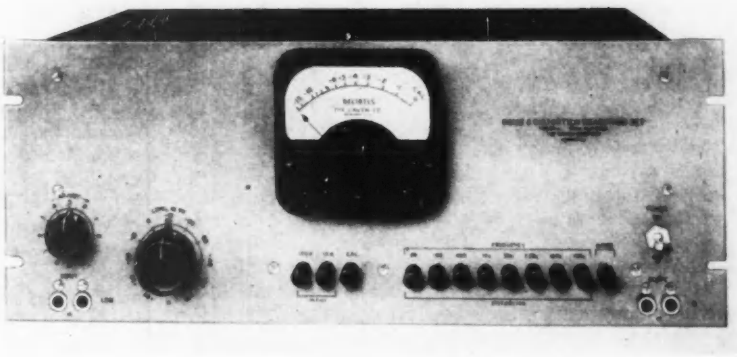
Potter Instrument Co., Inc., 115 Cutter Mill Rd., Great Neck, N. Y.



Circle No. 8 on Reader Service Card

CONTINUED ON PAGE 180

Push Buttons Tune Distortion-Noise Meter



This meter should not be confused with those employing null networks requiring careful frequency and phase adjustments at each reading. The fundamental circuit of the distortion and noise meter comprises a series of eight fixed band rejection filters covering the range 50 cycles to 15 kc, followed by a stable, high quality, wide range (50 cycles to 45 kc), high gain amplifier. There are no tube circuits or other sources of inherent distortion between the input and the filter network input, hence extremely low levels of distortion can be accurately measured over a wide level range. Each filter rejects uniformly a band ±3 percent of the indicated frequency, and passes uniformly the harmonics of this frequency, hence accurate frequency and phase adjustments are not necessary.

Two amplifier gain controls are provided, one an accurate step type covering the range +40 to —60 in steps of 10 db, the other a continuously variable control covering the range ±10 db. A calibrating voltage is provided internally for accurately adjusting the meter to read dbm.

The indicating meter covers the range 9 to —15 db, the range of the distortion meter is +40 to —60 db full scale meter reading or +40 to —75 db utilizing the meter scale. The range of noise and distortion that can be measured depends upon the level of the source being investigated. At a level of —15 dbm the limit is 60 db below or 0.1 percent distortion.

The output of the amplifier is protected by an electronic limiter which limits the output to full scale meter deflection. A pair of output jacks are provided for connecting an external scope in place of the output meter. A scope provides a simple means of determining the relative amplitudes of the harmonics or nature of the hum and noise indicated on the output meter.

Distortion Measurements: Filters provided for 50, 100, 400, 1,000 cycles, 5 kc, 7.5 kc, 10 kc and 15 kc with cut off of —70 db. Distortion measurements to 0.1 percent full scale meter deflection with zero level input.

Noise Measurements: With zero db input limit is —80 db. At +40 input limit is —115 db below input.

Amplifier Frequency Range: 50 cycles to 45 kc.

Level Range: +40 to —75 dbm. Instrument is a bridging type and levels are based on 1 mw into 600 ohms at zero db. Maximum input voltage is 77.5 v rms. Minimum input voltage can be as low as 0.00775 volts.

Accuracy: Filters are down 70 db at fundamental frequencies, and within 0.5 db of flat at the second harmonic. Absolute accuracy of measurement can be depended upon to be within ±5 percent.

Residual Distortion: No tube circuits or non linear devices between input of set and filter input.

Residual Noise Level: Below —80 db at gain control full on. Multiple gain control employed so that residual noise drops to —90 db when gain control is set at —30, —100 db when gain control is set at —20, etc.

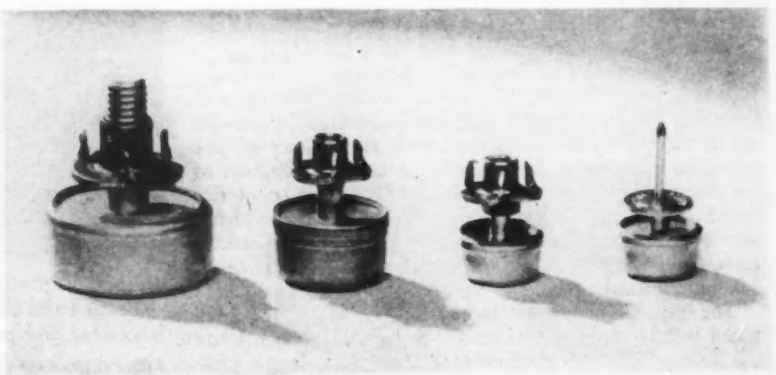
Input Impedance: 10,000 ohms balanced or unbalanced and 100,000 ohms balanced.

Meter: Large meter with single db scale, illuminated.

Daven Co., 191 Central Ave., Newark 4, N. J.

Circle No. 9 on Reader Service Card

Silicone Putty Glide Checks Wobble



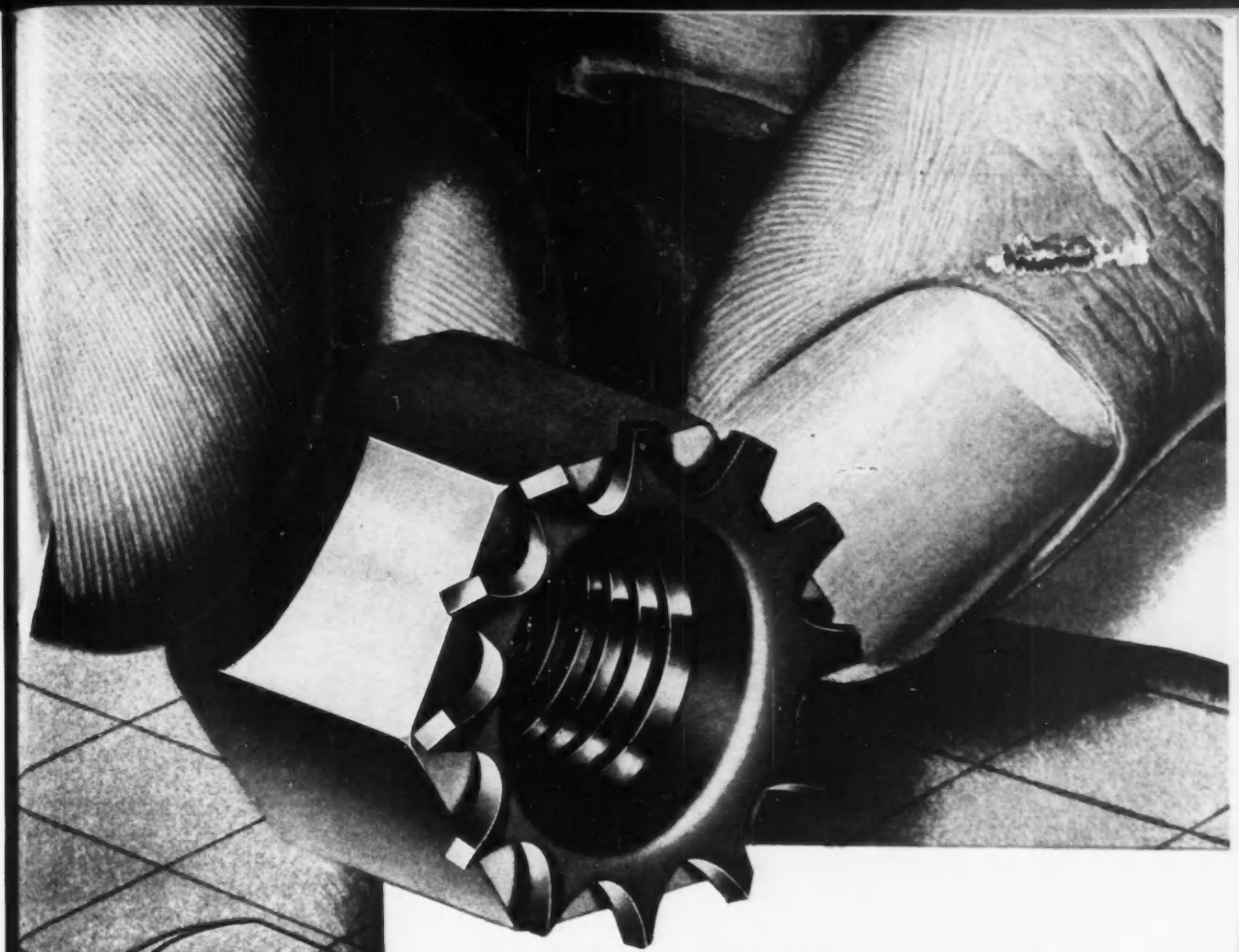
Wobble of cabinet legs due to uneven floors is automatically stopped by a self-adjusting glide. The device works on a simple hydraulic action, utilizing the properties of silicone putty to effect contraction and expansion in adjustment of the device to the uneven floor surface.

Regardless of where units equipped with the glides are moved they automatically readjust to the new location and will correct up to $\frac{5}{16}$ of an inch. They are supplied in sizes to withstand weights from 10 lb to 1 ton and can be supplied with a rubber body.

Ever-Level, Inc., New York 12, N. Y.

Circle No. 10 on Reader Service Card

CONTINUED ON PAGE 182



SHAKEPROOF *Announces*

KEPS

TRADE MARK

WRITE FOR FREE BOOKLET AND TESTING SAMPLES

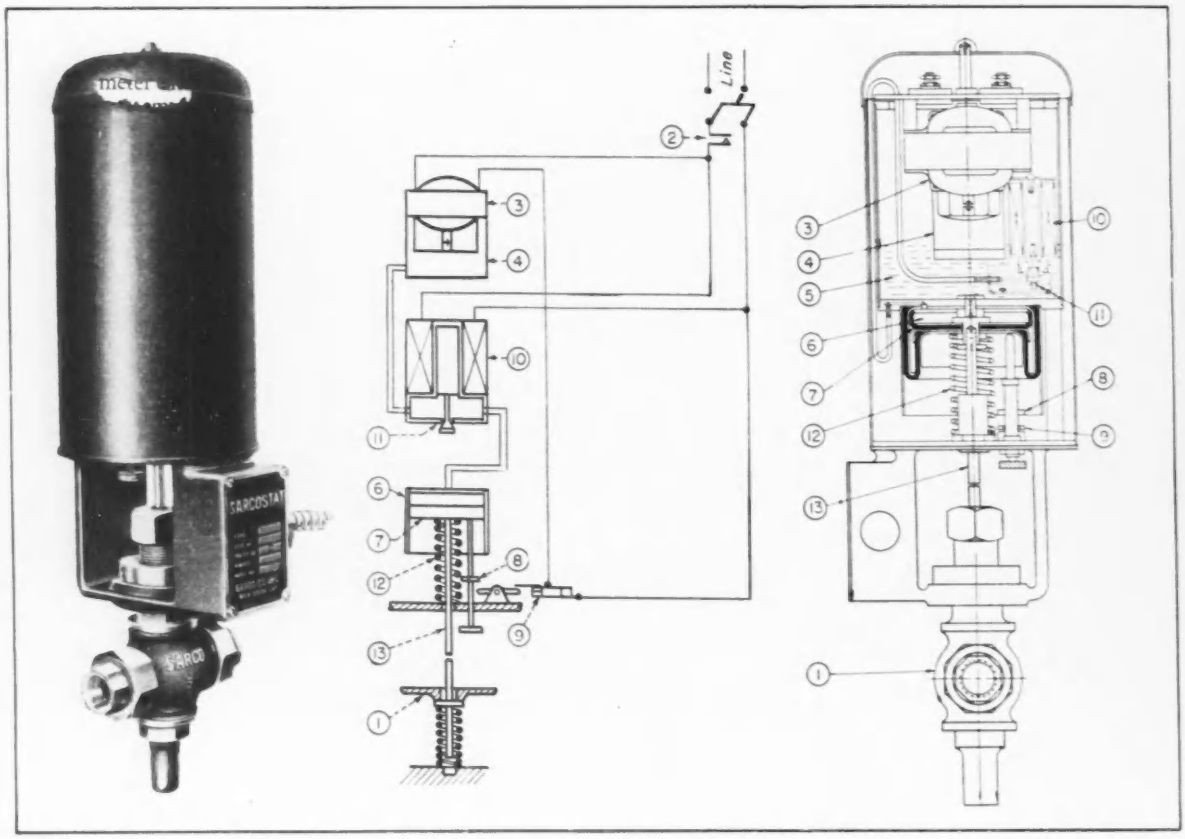
E NEW PATENTED FASTENING
RE-ASSEMBLED NUT AND
AKEPROOF LOCK WASHER

SHAKEPROOF inc

"Fastening Headquarters"

Division of Illinois Tool Works
2501 North Keeler Avenue, Chicago 39, Illinois
In Canada, Canada Illinois Tools Ltd., Toronto, Ont.

AMERICA'S GREAT RESOURCES PLUS A FREE ECONOMY MADE THIS BUSINESS POSSIBLE!



PHOTOGRAPH, FUNCTIONAL DIAGRAM AND CONSTRUCTION DRAWING showing the appearance and operation of valve. Operation can be regulated by thermostats or by liquid level or flow controls through 60 cycle 110 v current.

Solenoid Valve Needs Few Repairs

An electro-hydraulic motor valve for open-and-shut control operates two-wire, normally closed, for direct connection to 110 v a-c, 60 cycle current. It is designed for automatic operation by thermostats or pressurestats or by liquid level or flow controls.

As a shut-off valve in an inaccessible location, it can also be actuated by a push-button. The valve is constructed to handle steam, water, oil, gas, air, etc. It is also applicable for the remote control, by hand or thermostat, of steam or water branch mains for space heating systems. Gradual opening action protects the piping; quick closing prevents overheating.

The valve operator, actuated by hydraulic power, operates single seated valves up to 1½ in. or double seated valves up to 4 in. at 125 psi. It does this by direct thrust—without resort to pilots, gears or levers.

When the full stroke is used, standard valves will open in 60 sec and close in 10 sec. Also available with an adjustable stop for partial throttling.

The valve 1 is normally closed. Assuming the line switch to be closed, then the circuit will be made when switch 2 is closed. Switch 2 may be a push button switch for remote manual control, or an ordinary two-wire switch in a thermostat, pressurestat or other electric control element. Current flowing in the circuit now starts the motor 3, and at the same time closes valve 11 by means of small solenoid 10. The motor drives a rotary pump 4 which forces oil from sump 5 into cylinder 6 above the piston 7. The piston is thus forced downward and the valve 1 is opened.

When it reaches the wide open position, the adjustable plunger 8 opens the normally closed limit switch 9, which stops the motor 3. The valve 1 remains in the open position until thermostat switch 2 is opened, thus interrupting the current in the circuit.

Practically no current is consumed while the valve is open. When switch 2 opens, the solenoid 10 is de-energized, allowing needle valve 11 to open. The oil is thus allowed to flow back to sump 5, pressure under the piston is rapidly dissipated, and the valve 1 returns to the closed position, assisted by the spring 12.

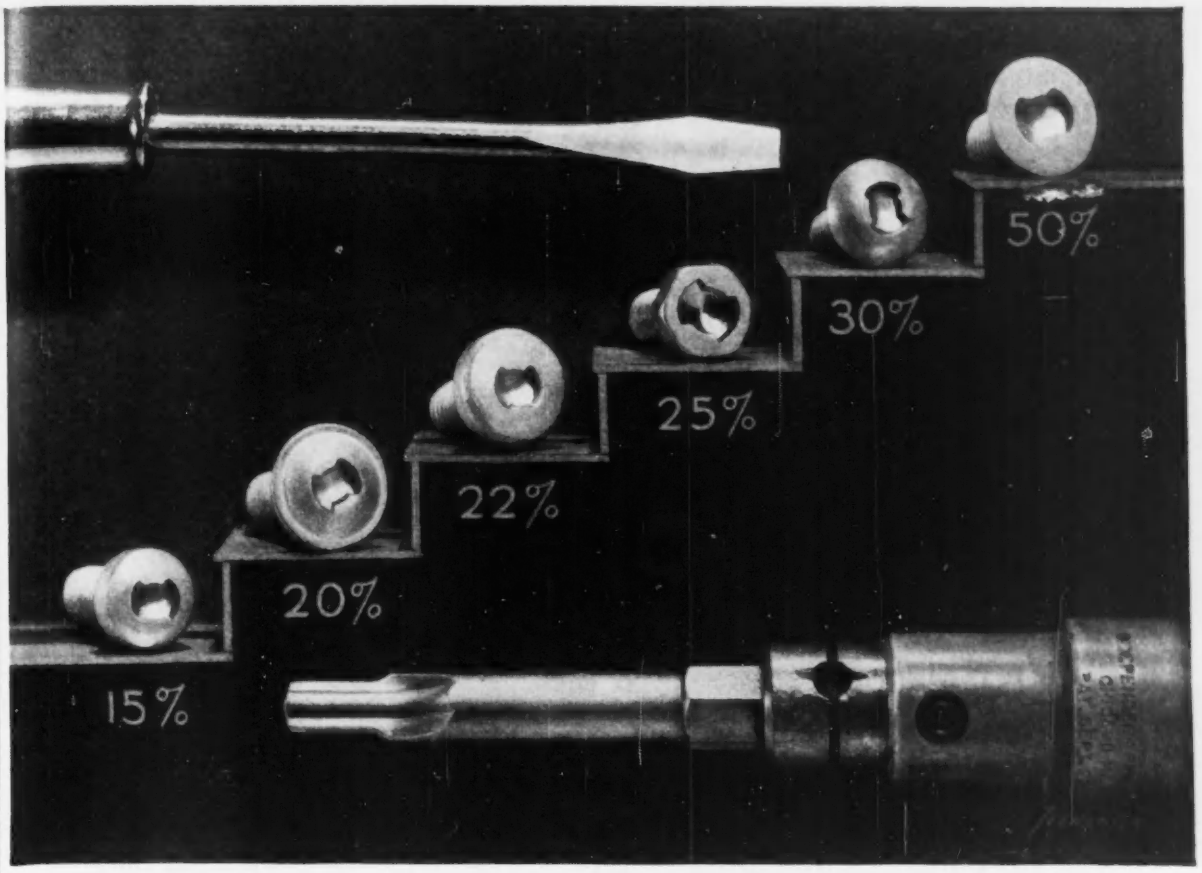
The motor used is of the shaded pole type, and develops approximately 1/100th hp at 2,200 rpm under full load. It is suitable only for direct connection to 110 v a-c single phase, 60 cycle current.

The pump will build up a pressure of 60 psi at full motor speed. The piston has a diameter of 2⅛ in. or an area of 6.213 sq in. Maximum stem thrust is 373 lb.

Sarco Co. Inc., Dept. 53, Empire State Bldg., New York, N. Y.

Circle No. 11 on Reader Service Card

CONTINUED ON PAGE 184



The Percentage Is All in Your Favor

These production increases are from the records of assembly lines where change-over was made from *other* types of *recessed head screws* to CLUTCH HEAD

HERE'S HOW CLUTCH HEAD PAYS OFF IN GAINS AND SAVINGS

High visibility of the recess cancels out slow-down hesitation . . . confident operators logically drive more screws.
Center Pivot entry prevents driver canting, insures straight driving . . . saves cost and delay of "fixing" burred or chewed-up heads.
Non-tapered driving engagement reduces skid hazard to zero . . . protects manpower and saves costly damage to materials.
All-square contact means no "ride-out" (as set up by tapered driving) . . . no fatiguing end pressure needed to combat "bucking."
The CLUTCH HEAD Lock-On hurdles "fumble spots" on the line . . . unites screw and bit as a unit for one-handed reaching and driving.
The rugged Type "A" Assembly Bit for unmatched tool economy . . . drives up to 214,000 screws non-stop on a high torque job.
Check the extra profit stemming from this steady production flow . . . no time-wasting stoppages for frequent tool changing.
Repeatable reconditioning multiplies the life of this bit . . . only a 60-second application of the end surface to a grinding wheel.
This simple operation restores original efficiency, *time and time again* . . . no expense or bother of back-to-the-factory shipments.
Consider the value of simplified field service . . . with a recess basically designed for common screwdriver operation.
Your service men and customers save time on field adjustments . . . any flat blade reasonably accurate in width will do.

These exclusive time and money-saving features are fully described and illustrated in the new CLUTCH HEAD *brochure. Send for your copy, mentioning the sizes and types of screws, standard or special, that interest you . . . to come by mail and without obligation.*

CLUTCH HEAD · SCREWS ·

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7

Meter Measures Flow Of Variety of Liquids

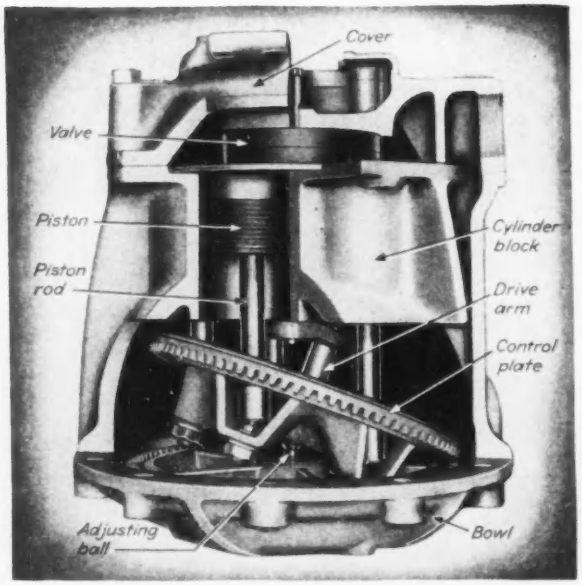
A new 2 in. meter for industrial liquids and petroleum products has been put on the market. The positive displacement meter has five vertical pistons and was developed to meet a demand for an accurate measurement, in the 110 gpm range, of a wide variety of liquids which could not be handled due to dryness or corrosiveness.

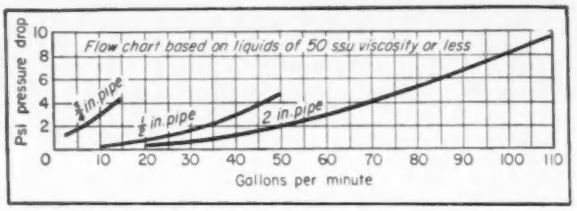
All interior parts are constructed of either special hard carbon or stainless steel to eliminate metal-to-metal contacts at friction points. The cylinder block and bowl is available in cast iron or bronze. Straight reading, predetermined quantity or ticket printing dials can be supplied.

All bearing and wearing surfaces are made of special hard carbon to resist wear, friction and corrosion. The plunger rods, valve port plate and drive arm are made of stainless steel. Parts are machined to close tolerances, so that a positive liquid seal is formed between the pistons and the cylinder walls and without metal to metal contact. All models are easy to adjust—the turning of one external screw lengthens or shortens the length of the piston stroke —displacing more or less liquid per cycle. To provide rigidity, the cover, cylinder block and bowl are made of cast materials with reinforcing ribs to withstand the pressures at which the various models are rated.

Meters are available in three sizes with capacities of 15, 50 and 110 gpm. A wide assortment of dials is offered.

Bowser Inc., Fort Wayne, Ind.





Circle No. 12 on Reader Service Card

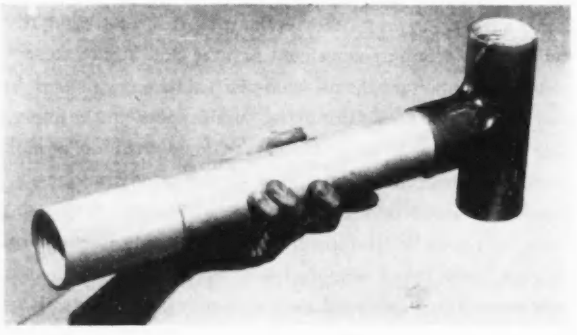
Fiber Material Can Replace Steel

A glass-fiber tube and pipe material can be used as a replacement for steel and other critical war metals in many commercial applications. It has the strength of steel and is rust and corrosion proof. It is now available as piping in the oil and chemical processing industry and as tubing for the building, electrical and allied fields.

The product is a laminated tubing in which glass fibers, in the form of cloth, mat or tape, are bonded with resins to develop a rugged tube impervious to extreme heat, chemical action and sledgehammer blows. Much of the research was done for the U. S. Navy, and it is already being used by the U. S. Army as well as in commercial applications.

Initial uses have been in the structural and electrical fields where it has found a ready acceptance for applications requiring a non-metallic product which possesses good electrical properties, is light in weight, and has a high strength factor.

Another field now being explored is the chemical processing industry. It has good resistance to many dilute and strong acids. It is possible that the product will replace corrosion-resisting materials which are more expensive, such as stainless steel, lead, and possibly unreinforced thermoplastic tubing of some types.



The oil industry offers many possibilities. The use of this new product in line and well pipe is indicated. The glass fiber is uneffected by most industrial and petroleum product liquids and gases. The fiber itself and the resin binder have enough resiliency to withstand shock loads imposed by fluid hammer and valve operation.

U. S. Plywood Corp., 55 W. 44th St., New York 18, N. Y.

Circle No. 13 on Reader Service Card

CONTINUED ON PAGE 18

MAXITORQ KEEPS GOOD COMPANY

"IN THE PACKAGING FIELD

Here's another "good company" friend of Maxitorq, the Pneumatic Scale Corp., that has selected our No. 23 Single Clutch as original equipment in their most ingenious Heat Seal Tea Bag Machines.

Automatically, at high speed, the bag material is fed from rolls, folded, crimped, filled with tea and sealed. Then tags are cut, attached to string, which is wrapped around the bag and stapled to prevent tangling. Such production requires a reliable, positive-acting clutch for smooth, uninterrupted operation.

The Maxitorq floating disc Clutch is made in 8 standard sizes from fractional to 15 H.P. at 100 r.p.m. Design is compact and streamlined . . . completely assembled on the body and shipped ready to slip onto a shaft. The floating disc feature (Separator Springs) prevents drag, abrasion and heating in neutral . . . and all Assembly, Adjustment and Take-apart are manual operations.

Carlyle Johnson engineers will provide practical solutions of your power distribution problems on request . . . and send you our new catalog covering single and double clutches, wet or dry . . . the automatic Overload Release Clutch and three styles of special driving cups.

Send for Catalog No. PE11

7CJ50

THE CARLYLE JOHNSON MACHINE COMPANY
MANCHESTER • CONNECTICUT

Sweep-Angle Valves Handle Slurry

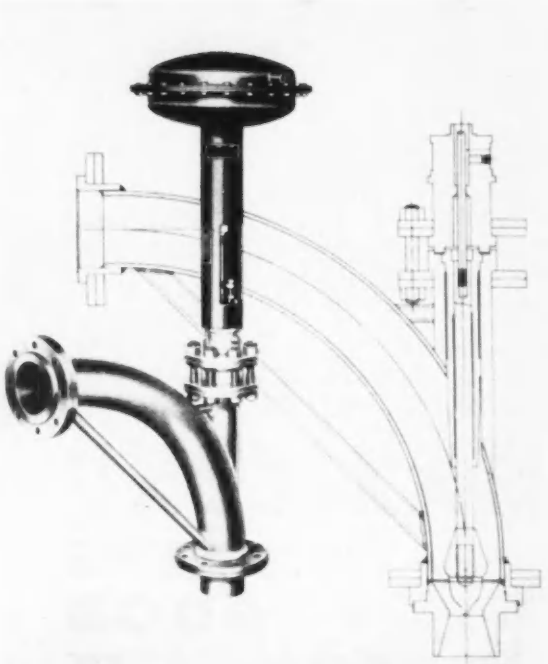
Long sweep angle valves are designed primarily to produce minimum turbulence in the flow through the valve and for use on slurry service where the solids in suspension tend to plug the conventional type control valves. Anything that can be pumped can be controlled by one valve.

The valve body is a long radius bend. It can be fabricated from any type pipe which can be bent and welded. Frequently, on chemical process applications special alloy piping must be used. Valves of the identical alloy are frequently difficult to obtain. The valve plug is streamlined to offer minimum flow resistance to fluid passing through the valve. The plug guide is extremely long to assure ample guiding of the valve plug even where it is operating very close to the seat. Valve plugs are of the high lift type, normal lift being ½ the nominal valve size, ie, a 6 in. valve will have 3 in. lift.

The seat is actually designed as a section of a venturi throat. This also assures minimum turbulence in the seat section. The venturi seat is built as one piece and is the retained type. Conventional seats are threaded and screwed into the valve body. Retained seats, however, have no threads, are lapped into the valve body and retained by the compression of the line bolting. There is no possibility of cutting out between the seat and the body, and the edge of the seat is exposed to atmosphere. Due to its single seated construction, tight shutoff can be provided. Due to the fact that the body has the same cross section as the connecting line, the maximum seat size is ½ the nominal pipe area.

Valve body bonnets will normally be provided of solid barstock. The bonnet is retained by through bolted construction. Oilite bearings on the stainless steel spring stem provide friction-free bearings regardless of whether the valve is inserted in a vertical or horizontal position.

Long sweep angle valves can be provided in any size from 2 to 6 in. Bodies can be supplied of carbon steel, type



304, 309, 316 and 347 stainless steel as standard. Any standard trim of the preceding stainless steels can be provided as well as stellited trim. The standard spring range of this type valve is 3-15 psi although higher spring ranges for high working pressures can be provided.

Hammel-Dahl Co., 243 Richmond St., Providence 3, R. I.

Circle No. 14 on Reader Service Card

Spectrum Analyzer Is Fully Shielded

The spectrum analyzer has been engineered carefully to avoid obsolescence, by making provisions for rapid interchange of RF heads that will provide coverage of the microwave spectrum to 30,000 mc. The unit features a functionally designed housing incorporating a 5 in. cathode-ray tube indicator, with a mounting that permits convenient use of a 35 mm camera for recording information.

The analyzer is a super-heterodyne receiver with a 5 in. cathode-ray oscilloscope output indicator. The local oscillator is frequency modulated by a sawtooth voltage wave which also produces a linear horizontal sweep on the cathode-ray tube. The spectrum analyzer displays the various frequency components of an input signal as vertical pips on the oscilloscope; the frequency and power of a component are indicated by its pip.

S-band and X-band heads are interchangeable in the basic equipment. Therefore, only the basic equipment and the two RF heads are required for the two bands. Additional RF heads, including wide range units, will be available later.

The new design also includes a stabilized IF amplifier for long term alignment. Potted transformers, oil filled condensers, and preferred type tubes are used to assure long life.

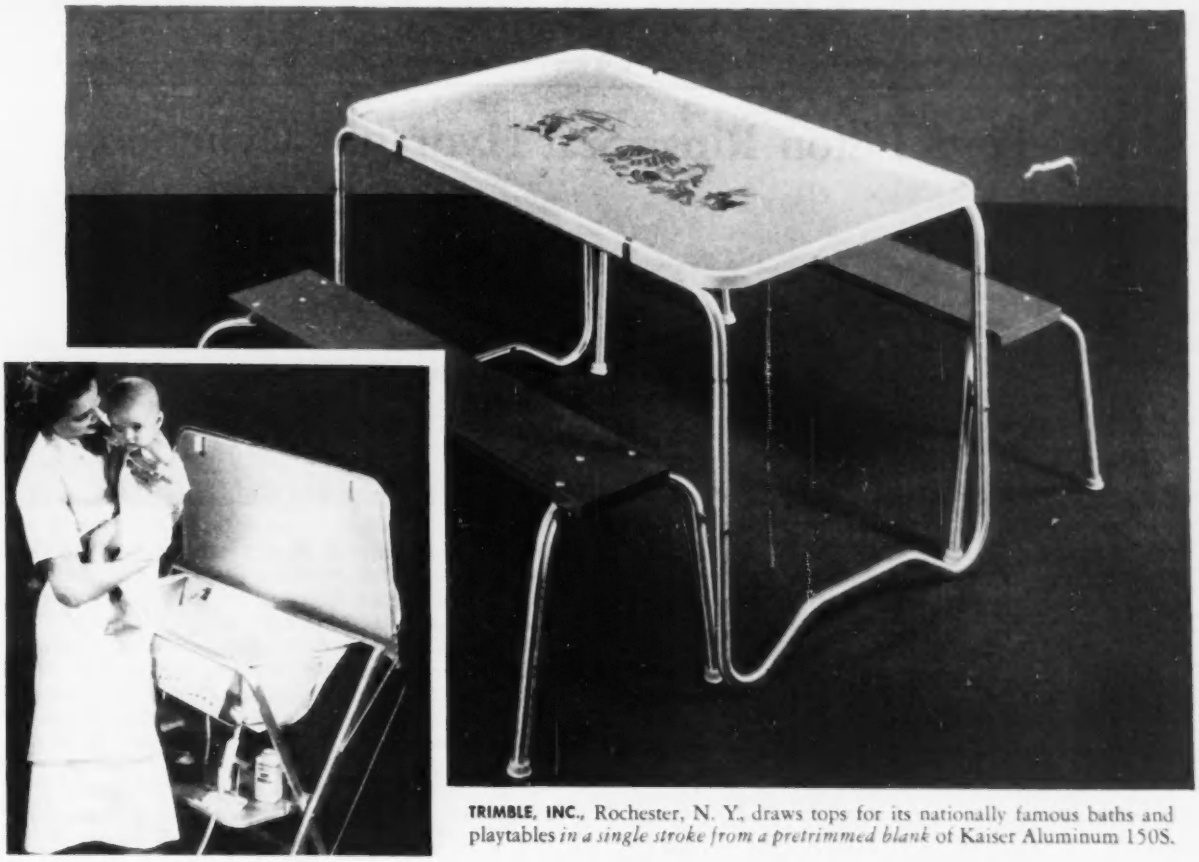
A high-voltage regulator unit, available as an accessory, can be mounted internally to furnish electronically-regulated voltages for RF heads incorporating wide range local oscillators or similar tubes as they become available.

Overall Gain, 130 decibels; IF Bandwidth, 50 kc; Sweep Frequency, 10 to 30 cps; Power Requirements, 115 or 230 v, 50-980 cps; Frequency Spread (max), 0.75 mc-in.; Frequency Range with S-Band head, 2,400 to 3,650 mcps; and Frequency Range with X-Band head, 8,500 to 9,660 mcps.

Robert A. Waters Inc., Dept. SA, 4 Gordon St., Waltham 54, Mass.

Circle No. 15 on Reader Service Card

CONTINUED ON PAGE 188



TRIMBLE, INC., Rochester, N. Y., draws tops for its nationally famous baths and playtables *in a single stroke from a pretrimmed blank* of Kaiser Aluminum 150S.

Trimble trims costs, improves products with new 150S

TRIMBLE, INC., previously produced their tray top, shown on the Playtable and Kiddie Bath above, from 52SH32 aluminum. Now, after a gauge for gauge conversion, they're using Kaiser Aluminum 150SH36—*and getting a better product at lower cost!*

Kaiser Aluminum 150S gives them increased dent resistance! Kaiser Aluminum 150S gives them a more attractive etched finish! Kaiser Aluminum 150S gives them the required strength!

And the lower material cost of Kaiser Aluminum 150S is a cash-in-hand advantage to Trimble—and scores of other manufacturers!

Why not let 150S—the world's most versatile aluminum alloy—go to work for *you?*

IF YOU'RE USING 3S, you'll get up to 40 per cent greater strength in finished parts by changing to 150S. Nearly the same workability. Harder surface. Finer grain structure. High lustre buffing. Superior etch finishes.

And up to 10 per cent less material cost where greater strength allows gauge reduction.

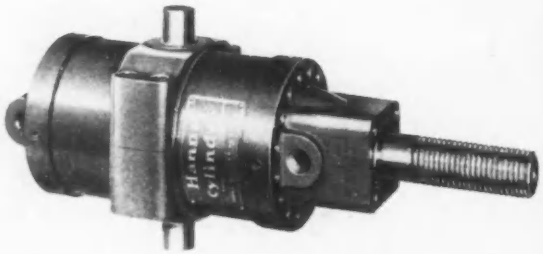
Give your product the advantages that only Kaiser Aluminum 150S can give it. Call the nearest Kaiser Aluminum office today for more information!

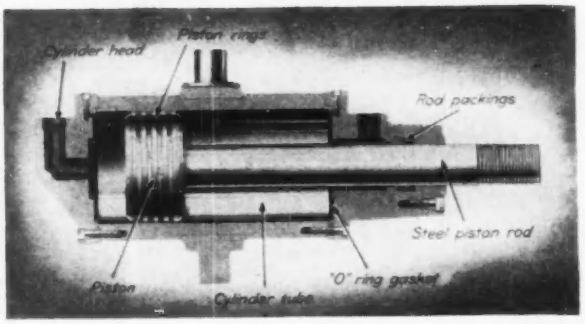
Kaiser Aluminum

A major producer in a growing industry

SOLD BY KAISER ALUMINUM & CHEMICAL SALES, INC., KAISER BUILDING, OAKLAND 12, CALIFORNIA . . . OFFICES IN:
Atlanta • Boston • Chicago • Cincinnati • Cleveland • Dallas • Denver • Detroit • Houston • Indianapolis • Kansas City • Los Angeles
Milwaukee • Minneapolis • New York • Oakland • Philadelphia • Portland, Ore. • Rochester, N. Y. • Seattle • Spokane • St. Louis
Wichita • EXPORT OFFICE, OAKLAND, CALIFORNIA • WAREHOUSE DISTRIBUTORS IN PRINCIPAL CITIES

Four Piston Rings Seal Hydraulic Cylinders





Hydraulic actuating cylinders are designed to give efficient push, pull, lift, press, clamp or control power on working pressures up to 1,500 psi. Bore sizes range from 1½ to 10 in. Cylinders can be mounted in 8 standard styles. All meet JIC standards.

Cylinder heads can be rotated for varied part location. Large area of metal-to-metal contact between cylinder head and tube prevents cocking and assures squareness of mounting surfaces to cylinder bore. A three-way combination of true round piston rings with lapped sides, precision-cut ring grooves and mirror-honed round tube assures minimum oil slip. Cylinder tube is accurately bored and honed centrifugally cast Meahanite. Close grain and high finish give low friction and minimum leakage past piston. There are four air vents to each cylinder head. Air can be vented regardless of mounting position. Rod packings are spring backed to assure correct compression for maximum sealing efficiency and low friction.

Heat-treated alloy socket head screws for maximum strength are countersunk into heads to give a clean-cut appearance. Steel piston is finish machined after assembly to piston rod for concentricity. Piston ends are relieved to permit piston rings to act as scrapers for protection against scoring. Self-sealing gaskets of the "O" ring design are efficient under all operating conditions. Piston rod ends can be machined to meet requirements of application. Male thread standard unless otherwise specified.

Hanna Engineering Wks., 1765 Elston Ave., Chicago 22, Ill.

Circle No. 16 on Reader Service Card

Oscillograph Provides Internal Calibration

In a recently announced quantitative, 10 megacycle cathode-ray oscillograph, time calibration is provided for the horizontal sweep of the instrument and regulated voltage calibration for vertical deflection. Both time and voltage calibration are accomplished by substituting a calibrating signal for the input signal.

The Y axis of the instrument, which includes a fixed signal-delay line, provides a sensitivity of 0.1 volt peak-to-peak per inch. The frequency response is down 3 db at 10 megacycles with no positive slope in the high-frequency range. Response falls off slowly past the 10 megacycle point, so that the instrument is usable at frequencies considerably higher than 10 megacycles. The instrument will synchronize on signals higher than 15 megacycles.

Transient response of the Y axis is such that a pulse having a rise time of 0.01 microsecond or faster will be reproduced with a rise time not exceeding 0.03 microsecond. No high-frequency overshoot is observed.

Both driven and recurrent sweeps are variable continuously from 0.1 second to 5 microseconds. Sweep length may be expanded to the equivalent of 30 in., effectively producing a sweep speed of six in. per microsecond, without introducing non-linearity.

Instrument Div., Allen B. DuMont Laboratories, Clifton, N. J.

Circle No. 17 on Reader Service Card

CONTINUED ON PAGE 189

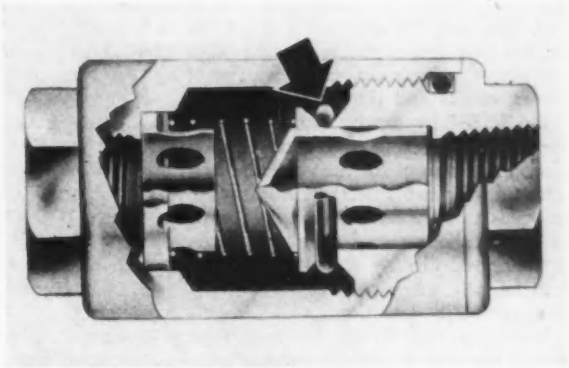
"O" Ring Seals Standard Check Valve

Circular seal check valves are being made available as standard items in a full range of sizes from ⅛ in. to 1 in. NPT. Two models are available designed for standard and very low pressure systems; both are poppet type using an "O" ring seal. Absolute sealing, minimum flow resistance, and leakproof dependability for pneumatic and hydraulic systems are claimed.

Check valves are bubble-tight in leakage tests, according to the manufacturer, providing absolute sealing at any differential pressure. An added feature of the design is quick opening and positive closing at extremely low pressures. Opening pressures are as low as 1 psi differential in the standard pressure types and 4 in. water column in the low pressure types. Poppets close before return flow starts.

The operating principle utilizes a synthetic rubber "O" ring between conical surfaces on the poppet and valve housing. As the line pressure works against the outer periphery of the "O" ring, increased line pressure greatly increases the sealing pressure. The circular seal provides only the sealing; the poppet load is carried by a metal shoulder in the valve housing. In the open position, the fluid flows around the streamlined poppet, giving the check valve excellent flow characteristics with low pressure drop. The spring retainer, which serves as a poppet stop, has ample ports to allow full flow even under surge conditions. The valves have the advantages of clean design, positive action, long life, minimum maintenance, negligible flow resistance, plus leakproof performance, providing in one check valve all of the elements considered of primary importance in modern concepts of pneumatics and hydraulics. The wide range of adaptability permits the user to make installations in all pressure ranges without turning to a variety of check valve types and styles.

Maintenance economies are claimed due to the fact that the resilient synthetic rubber "O" ring absorbs shock and virtually eliminates wear or misalignment of the poppet and body. The pulsating action of shock waves and hammer



is absorbed by minor contraction and expansion of the "O" ring, preventing chatter or pound in the poppet. Any abrasive particles which come between the poppet and the seat are pushed aside by the "O" ring, maintaining a tight seal without damage to the poppet or the seat. Where unusually abrasive materials have passed through the valve, replacement of the "O" ring will normally restore the valve to correct operating procedure. Since the "O" ring cushions the load, it is not necessary to use special wear-resistant materials in the poppet and seat. Using the same materials throughout eliminates the problems of electrolytic action and differing temperature expansion for the valve parts.

Body and poppet are brass or aluminum in standard model. Circular seal is Buna N. Standard pressure is 3,000 psi, with 0 to 1 psi covered in low pressure type. Nearly all industrial fluids can be handled with suitable parts. Opening pressure is 1 to 5 psi in standard valve, 4 to 8 in. of water in low pressure unit. In both valves the poppet closes before the start of return flow. Sizes from ⅛ to 1 in. are available.

James-Pond-Clark, Pasadena, California

Circle No. 18 on Reader Service Card

Mass Spectrometer Analyzes Gases

A mass spectrometer tube aids scientists in the analysis of chemical compounds, especially gases. The spectrometer tube, known as an ion resonance mass spectrometer tube, is expected to be helpful in the many fields of measurements. High-precision measurements can be made in a number of special analysis problems.

In the ion resonance tube, when in operation, electrons from the filament are accelerated and form ions which are in turn accelerated in a space where crossed magnetic and RF electric fields are maintained. In these crossed fields, ions describe spiral paths. Those ions with the proper mass to resonate with the crossed fields will gain energy and describe larger and larger orbits until they reach the collector and are measured.

The ion resonance tube has a sensitivity comparable with conventional spectrometers. The resolution of the tube is high at low masses and it is expected that resolution up to mass 100 can be achieved. Approximately 150 v d-c and RF voltages of about 1 volt at frequencies up to 5 megacycles are employed. The filament utilizes 5 amp at 15 watts. The tube has a glass envelope and requires a magnet with a 1 in. gap and a field strength of several thousand gauss. The tube can be mounted in any position.

The extreme sensitivity of the tube permits operation with small samples. A wide range of operating pressures are possible; resonance peaks have been obtained at pressures of 10-7 millimeters of mercury. It is felt that new fields of research may be opened to accurate investigation.

General Electric Co., Schenectady 5, N. Y.

Circle No. 19 on Reader Service Card

CONTINUED ON PAGE 190

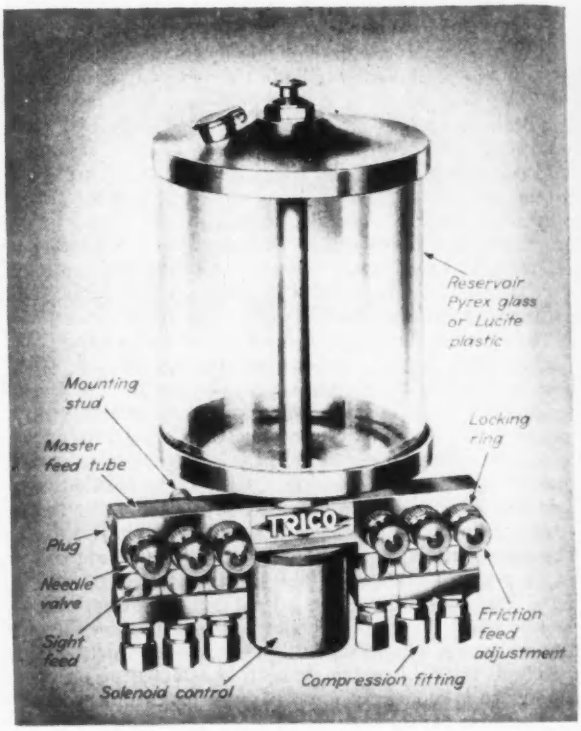
Solenoid Controls Central Lubricating System

A centralized system lubricates 1 to 20 bearing points on machine tools, punch presses, high speed production equipment, automatic machines, printing presses, conveyors, chains and similar applications.

Oil is fed by gravity from large reservoir through individually adjustable needle-valves and through copper or Neoprene tubing to the various bearings. When the solenoid, located directly below reservoir, is energized, oil feeds to all lines. Feeding stops when current is shut off. It can be operated from any convenient current outlet or connected directly across motor starting switch to feed automatically when starting, or stop feeding when equipment is not in operation.

Manual control type has a shut-off lever at top. Supply of oil remains in master tube at a constant level with needle-valve port holes for feeding when starting. Sight feed fittings show the amount of oil being fed to bearings. Needle-valve assembly is ¼ in. above base of master feed tube to provide a sediment chamber where all foreign matter is trapped. Individual feed lines are provided with vibration and leakproof friction-fit needle-valve adjustments, permitting independent regulation. Locking arrangement prevents change of adjustments due to vibration or tampering.

Oil reservoirs and sight-feed fittings are available in Pyrex glass or shatterproof Lucite. Reservoirs are available in three sizes: 1 pint, 1 quart and 2½ quarts. With the exception of precision-built, machined steel needle-valve, polished steel filler cap and stainless steel solenoid with cold rolled nickel plated housing, all metal parts are of machined brass. Solenoids are of standard voltage—115 and 230 v, 60 cycle—110 and 220 v, 50 cycle.



Trico Fuse Mfg. Co., 2948 N. 50th St., Milwaukee 12, Wis.

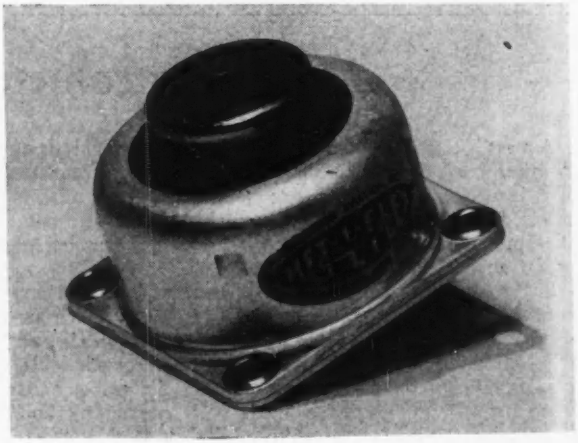
Circle No. 20 on Reader Service Card

Vibration Mount Is All Metal

A cup type all-metal, all-weather mount for aircraft and general industrial use comes in four sizes: 2½-6lb; 5½-12½ lb; 10-20 lb; and 18-40 lb. It is all-metal and incorporates an all-steel, resilient material impervious to temperature.

The cup-type mount provides performance and balanced equipment protection for sub-base mounting systems. This unit also provides vibration isolation efficiency exceeding government performance specifications, combined with excellent shock absorbing characteristics. An auxiliary element located in the base of the mount cushions overloads arising from landing shock, gunfire recoil and combat maneuvers. One series have the structural strength required to meet the 30 "G" drop-test requirements of Specification AN-E 19.

A second series offer uniformity of operation from —90 C to 175 C. Non-linear deflection characteristics provide adequate load range and effectively combine vibration isolation and shock protection. Interwire friction reduces amplification at resonance and effects greater stability in the mounting system.

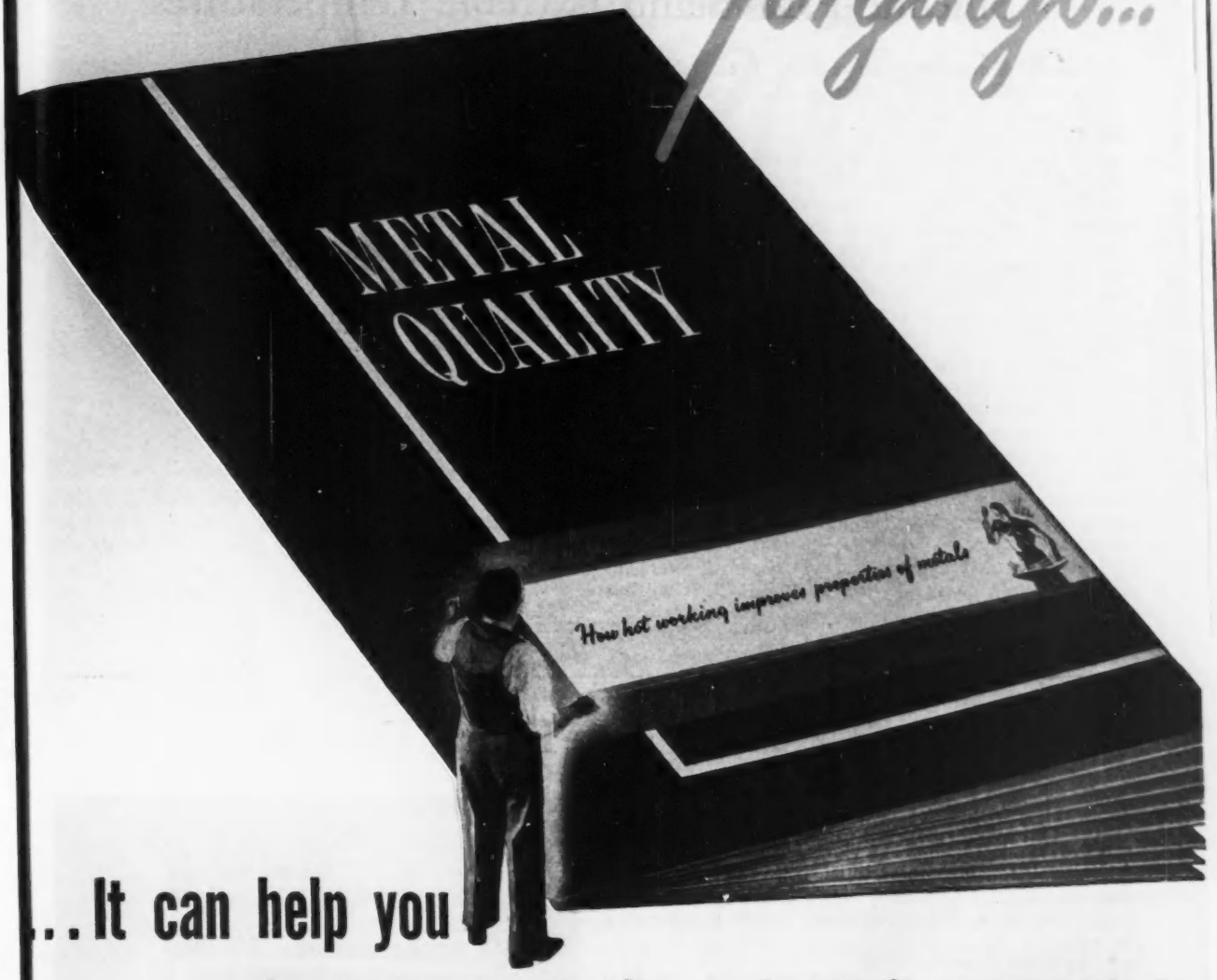


Robinson Aviation, Inc., Teterboro, N. J.

Circle No. 21 on Reader Service Card

CONTINUED ON PAGE 192

This REFERENCE BOOK on *Forgings...*



...It can help you reduce cost at the point of assembly!

There are thousands of instances where forgings have reduced the cost of parts at the point of assembly. This booklet containing 60 pages of authoritative information on forgings formed to close tolerances by the use of closed impression dies tells how cost of reductions may be obtained. Forging production techniques are described and illustrated.

Forgings provide rapid assembly of complex parts by welding adaptability of widest range; forgings permit reduction of dead weight because maximum strength and toughness are obtainable in lighter sectional thicknesses; forging to shape avoids waste of metal and reduces machining and finishing time-cost. The metal quality and cost-reducing possibilities obtainable in forgings cannot be equalled or duplicated.

Recheck every stressed part in your equipment, as well as simple handles and levers, and consult a forging engineer about possibilities for reducing costs at the point of assembly. Only a forging engineer can inform you fully regarding the numerous advantages obtainable with forgings. Your copy of this reference booklet on forgings is available now. Attach coupon to your business letterhead.

DROP FORGING ASSOCIATION
605 HANNA BUILDING • CLEVELAND 15, OHIO

DROP FORGING ASSOCIATION • 605 Hanna Building • Cleveland 15, Ohio

Please send 60-page booklet entitled "Metal Quality—How Hot Working Improves Properties of Metals" 1949 Edition.

Name ..... Position .....

Company .....

Address .....

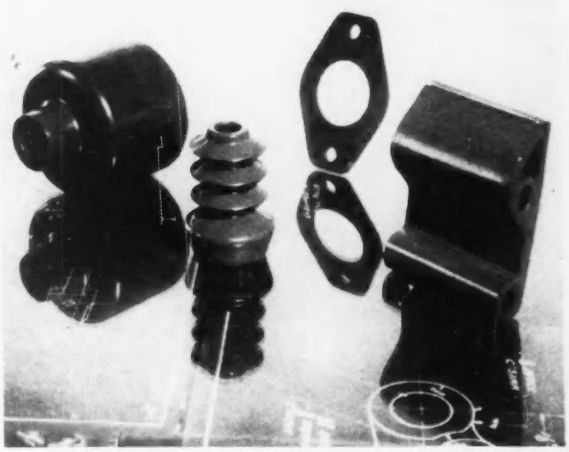
Silicone Parts Stand Extreme Temperatures

Gaskets, sleeves, washers, channel, seals, tubing and other molded, extruded, punched and lathe or die-cut rubber parts can be furnished in quantity, fabricated from the new silicone 161 rubber compound. This new compound can be fabricated more quickly and easily than previously developed silicone rubbers and the cost of production parts has been considerably reduced. The new silicone is red in color, which eliminates possible soiling during end fabrication. While the dielectric strength of this new compound is only 170 volts/mil, it retains all the other outstanding qualities of silicone rubbers to offer maximum resistance to extreme temperatures, weathering, chemicals, and oils.

Silicone 161 rubber parts have a tensile strength of from 470 to 520 lb per square inch, elongation of 110 to 160 percent, and a high specific gravity of 2.1.

Parts feature a wide applicational value in installations where extreme temperatures are encountered, such as in industrial ovens, steam generators, jet aircraft engines and die-casting machines. In addition, the compound is recommended for use in systems handling hot or cold fluids, gases, and vapors.

It will not become brittle and will not crack after long exposure to air, ultra-violet rays or ozone. Parts will not turn to carbon despite extreme heat, and they can be used to replace metal constructions which have been forced upon design engineers because of limited thermal stability of organic rubber.



Stalwart Rubber Co., 166 Northfield Rd., Bedford, Ohio

Circle No. 22 on Reader Service Card

Sensitive Relay Carries Heavy Duty Rating

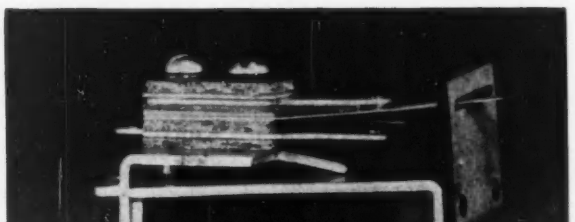
The coil of this sensitive relay is 15,000 ohms and operates on 5 milli-amperes d-c. Contacts are snap action and will handle 15 amp, 115 v or 7½ amp, 230 v, a-c.

Designed for high differential between pull in and drop out the relay is normally high speed in action. However, it may be used for time delays from a fraction of a second up to thirty seconds or more by the use of capacitors connected across the field.

Due to the snap action there is no possibility for the contacts or armature to float between open and closed positions. Even in cases where the field changes at a very slow rate the contacts have high pressure at all times.

Standard coil is 15,000 ohms. Contacts are Single Pole Double Throw or Double Pole Double Throw. Other coil and contact combinations can be furnished.

Coil is thoroughly impregnated against moisture. The relay may be mounted in any position. Overall dimensions are 1¼ in. wide, 2½ in. long, 1¾ in. high.



Assembly Products, Inc., Chagrin Falls, Ohio

Circle No. 23 on Reader Service Card

WIRE-WOUND AND CARBON CONTROLS are combined in concentric unit to give independent control in either circuit on TV applications. Clarostat Mfg. Co., Dover, N. H.

Circle No. 24 on Reader Service Card

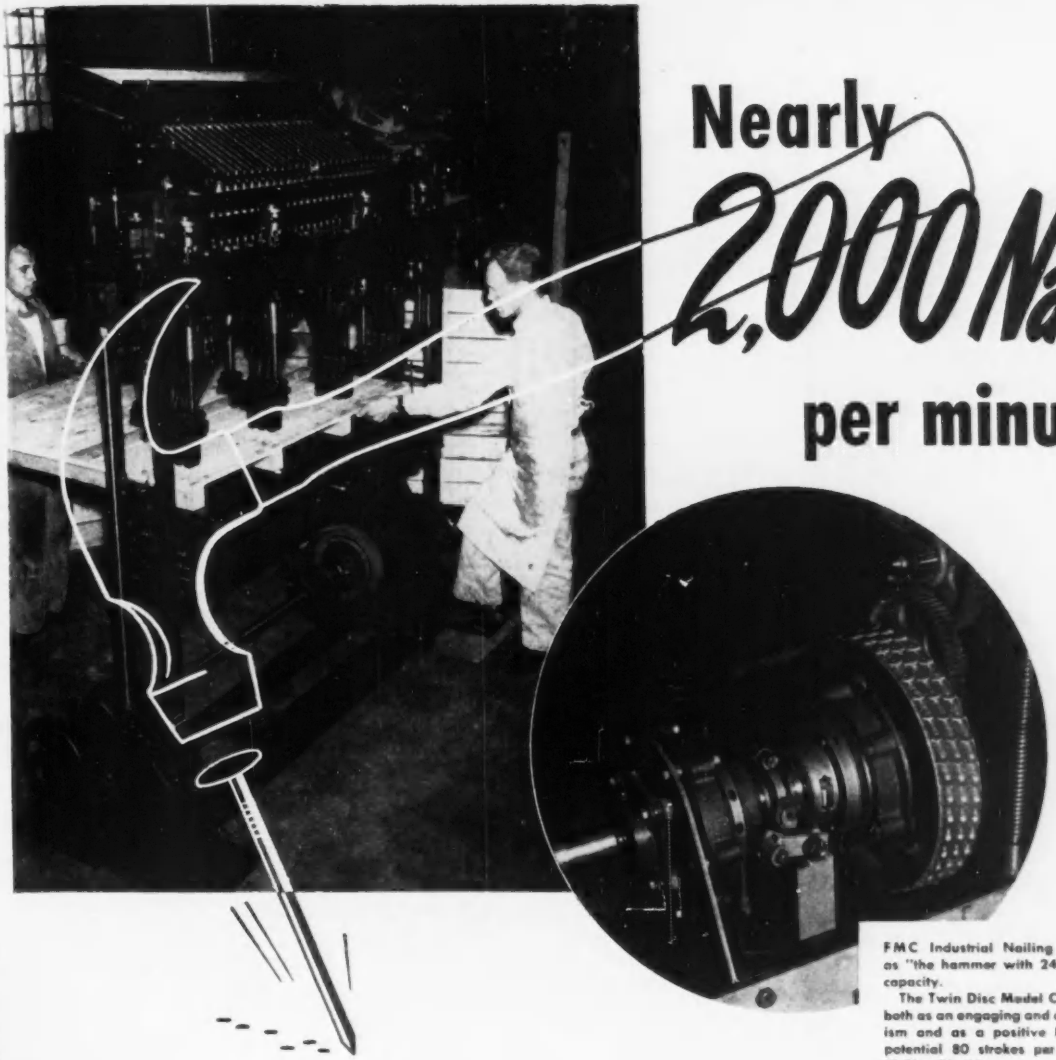
UNIVERSAL JOINTS, yoke type, are heat treated alloy steel, ground on bearing surfaces. Meet aircraft and shipbuilding specifications. Boston Gear Wks., Quincy, Mass.

Circle No. 25 on Reader Service Card

EDGE GUIDE for paper and textiles is 3½x4x8 inches. Photoelectric cell keeps rolls running true. Accuracies of 0.03 and 0.005. Airlectron Inc., 110 Washington St., New York, N. Y.

Circle No. 26 on Reader Service Card

CONTINUED ON PAGE 194



Nearly 2,000 Nails per minute

FMC Industrial Nailing Machine is known as "the hammer with 24 heads" for its high capacity.

The Twin Disc Model CL Duplex Clutch acts both as an engaging and disengaging mechanism and as a positive brake to control the potential 80 strokes per minute cycle of the Nailing Machine.

This hammer has 24 heads—geared to operate at 80 strokes a minute. *Every* stroke is an operator-actuated connect and disconnect. At the rate of four strokes every three seconds, the punishment the *clutch* must handle is nothing short of amazing.

That's why on its latest model Industrial Nailing Machine, the Food Machinery and Chemical Corporation selected a Twin Disc Model CL Duplex Clutch.

This clutch, with its positive action, is one reason why the Nailing Machine is rated with a high capacity. For the nail chucks and driving assembly are released from the nailed object immediately after each nailing stroke, enabling the operator to index the work through the machine for successive nailing operations.

For assured customer satisfaction on the machines you design and build, always specify Twin Disc Clutches, the result of 32 years of specialization in the design, manufacture and application of power transmission units.

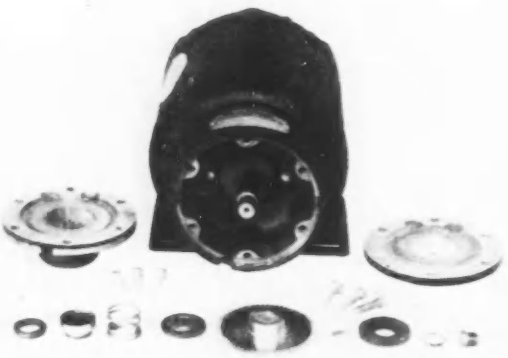
TWIN DISC
CLUTCHES AND HYDRAULIC DRIVES

Clutches & Hydraulic Drives

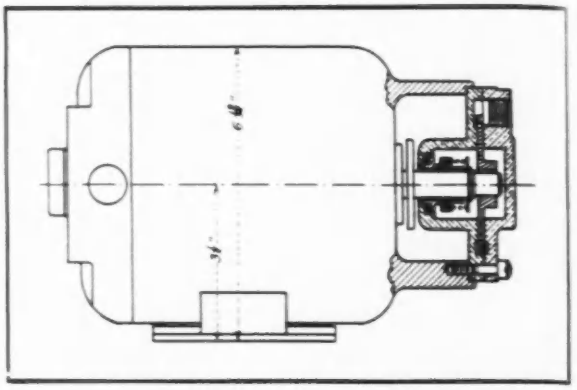
TWIN DISC CLUTCH COMPANY, Racine, Wisconsin • HYDRAULIC DIVISION, Rockford, Illinois

BRANCHES: CLEVELAND • DALLAS • DETROIT • LOS ANGELES • NEWARK • NEW ORLEANS • SEATTLE • TULSA

Turbine-Type Pump Occupies Little Space



Multi-vaned, shaft-mounted impeller is chrome-plated bronze and turns with no metal-to-metal contact in the raceway. Maximum working pressure is 160 psi with cold water.

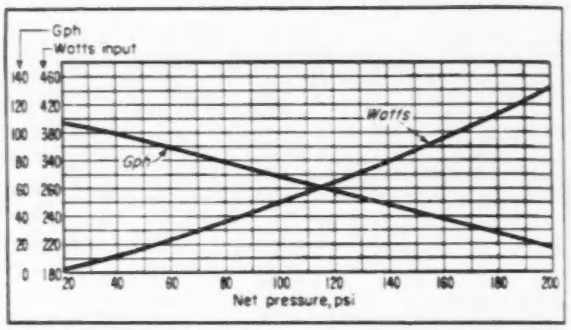


No relief valve is required and there is a definite shut-off pressure. The rotor is in hydraulic balance, without end-thrust in either direction. A mechanical seal closes over the shaft.

An overall 10¾ in. length by 6 19/32 in. height permits compact installation of pump. Applications include water-cooled radio tube assemblies, aqueous spray installations and small capacity, high pressure boiler feed service.

The impeller turns with no metal-to-metal contact within the matched raceway formed by the 85-5-5 H5 bronze pump body and pump cover. Hydraulic balance prevents end-thrust in either direction. A mechanical seal is fitted to the single stainless steel pump-motor shaft. The motor is ¾ hp, 1 ph, 60 cy, 110 v, 3,450 rpm, capacitor high torque with automatic overload protection. The components are easily disassembled and replaced, and the simplified design minimizes maintenance. Operation is claimed to be extremely quiet. Discharge is nonpulsating. All parts in contact with liquid are corrosion-resistant.

A wide range of operating pressures is possible. The performance curve indicates efficient operation up to a recommended maximum working pressure of 160 lb with clear cold water. Present uses involving working pressures at 120 lb show excellent life expectancy and economy.



Wayne Home Equipment Co., Inc., Fort Wayne, Ind.

Circle No. 27 on Reader Service Card

D-C Millivoltmeter Has High Sensitivity

Characterized by an extremely low internal residual, high-input impedance, and gain stability comparable to that obtained in a-c vacuum tube voltmeters, the instrument can detect d-c voltages as low as 5 microvolts, or with suitable shunts, currents to a level of 10-11 amps.

The output from the instrument is an a-c voltage, the rms magnitude of which is precisely 1,000 times that of the d-c input. The transfer gain is stabilized by precision components and a strong local feedback loop.

There is no drift in the instrument. No zero set or balance controls, or calibration checks during measurement are necessary. A dynamic range of 10,000 to 1, and linearity of the output-input proportion are other advantageous features. An attenuator is provided at the input to extend the range of the unit by up to 10 v d-c. The instrument is operated from the 115 v 60 cycle line.

In addition to the field of low level d-c measurements, it has important application for computers and for servomechanisms. Many times the output of some d-c computer is a voltage too small to be processed by conventional d-c amplifiers. This model can be used to convert this signal to an amplified a-c voltage that may then be processed by carrier frequency networks. It can be used as part of a controller in a servo which must establish functional correspondence between a low level, high impedance signal, and a shaft position.

Industrial Control Co., 1462 Undercliff Ave., New York 52, N. Y.

Circle No. 28 on Reader Service Card

CONTINUED ON PAGE 196

THE HOUSEWARES AND ACCESSORIES FIELD

Our part in the production of contract requirements for this broadly varied field has been a major one for many years. We take particular pride in our ability to satisfy the many needs of so large and diverse a market.

ONE OF THE MANY WE SERVE

WITH ALUMINUM PARTS

Whatever your product may be, if it requires aluminum parts, our diversified skills and facilities can produce them. Alloys and thicknesses? What are your needs? We custom roll sheet metal to your parts specifications, in our own modern rolling mill. Finishes? Satin, color, anodic, polished or chemically brightened. Engineering? Design? Our staffs are at your service.

Your inquiry will focus our full attention on *your* needs . . . at once.

CONTRACT DIVISION

ALSO MAKERS OF
MIRRO
The Finest Aluminum Cooking Utensils

ALUMINUM GOODS Manufacturing Company

MANITOWOC WISCONSIN

FIFTH AVENUE BLDG., NEW YORK 10 MERCHANDISE MART, CHICAGO 54

Air Filter Controls Line Pressure

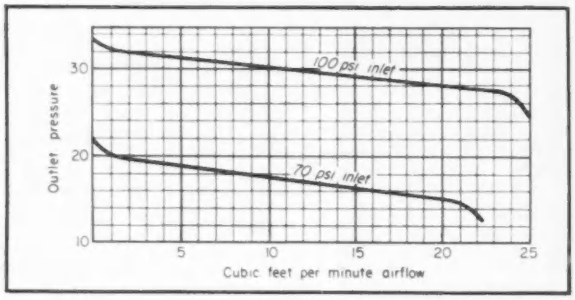
An air reducing valve and an air filter are combined in this unit. It is compact and easily mounted and serviced. Capacity averages 20 to 25 cfm at 75 psi inlet pressure.

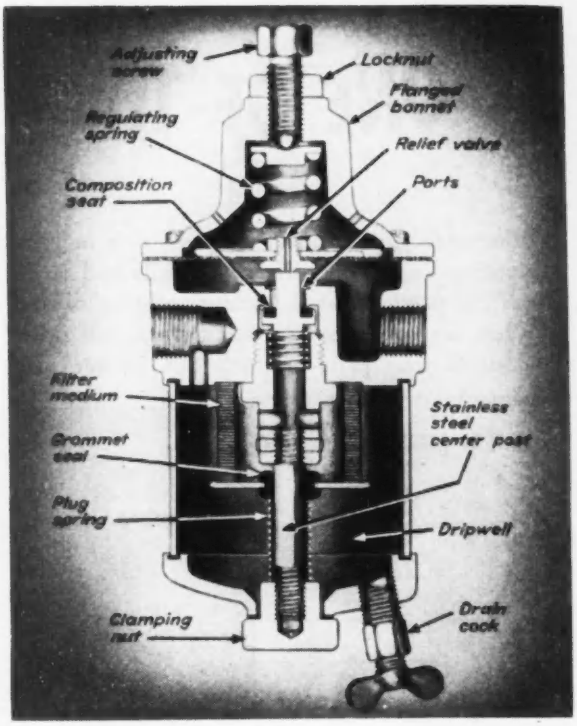
The filter is a cylinder, made of phenolic resin-impregnated cellulose ribbon, capable of removing 40 micron particles. The filter area is 5 sq in.

The regulator is controlled by an adjustable spring. A Neoprene diaphragm, nylon reinforced, seals the regulator head. This diaphragm can be replaced by unbolting the flanged bonnet. The pressure-regulating spring is tempered steel; the plug spring is beryllium copper.

One $\frac{1}{4}$ in. inlet and three $\frac{1}{4}$ in. outlets are spaced at 90 deg intervals around the housing. The four openings permit the mounting of a pressure gage or multiple outlets. Regulators are offered in 0-30, 0-60, and 0-125 psi ranges, for inlet pressures up to 150 psi.

Conoflow Corp., 2100 Arch St., Philadelphia 3, Pa.





Circle No. 29 on Reader Service Card

OVERCURRENT RELAY has extremely inverse current-time characteristics. Burden is 0.3 va at maximum current. Used where long time delay is required for light overloads. Westinghouse Electric Corp., Box 2099, Pittsburgh 30, Pa.

Circle No. 30 on Reader Service Card

DIAPHRAGM VALVE operates on pressure differentials below 75 psi. Sizes $\frac{1}{2}$ to 2 inches. Standard and packless valve bodies are available. Regulates steam, water, oil or gases. Powers Regulator Co., 2720 Greenview Ave., Chicago 14, Ill.

Circle No. 31 on Reader Service Card

RELAYS OFFER high contact pressure and coil power to withstand shock and vibration. Redesigned model of standard relay. R-B-M Div., Essex Wire Corp., Logansport, Ind.

Circle No. 32 on Reader Service Card

CONTROL STATIONS with 12 amp 600 v, 60 amps 110 v a-c capacities are offered in 12 unit, oil-tight enclosures. Push buttons come in combinations of 1 to 12 units. Westinghouse Electric Corp., Box 2099, Pittsburgh 30, Pa.

Circle No. 33 on Reader Service Card

SPEED REDUCERS save space in straight line $\frac{3}{4}$ to 1 hp applications. Torque 1,000 in-lb, speed 1,800 rpm maximum. Ball bearings used throughout. Universal Gear Corp., 19th and Martindale, Indianapolis 7, Ind.

Circle No. 34 on Reader Service Card

LARGE DIAMETER CYLINDERS for 125 and 160 psi have been redesigned in 14 and 16 in. sizes. Connections are 1 and $1\frac{1}{4}$ inches respectively. For air, oil or water. Hannifin Corp., 1126 S. Kilbourn Ave., Chicago 24, Ill.

Circle No. 35 on Reader Service Card

SWITCHBOARD INSTRUMENTS are designed to permit shadow-free illumination of scales with room lighting. Glass protrudes; scale is flush with case front. Meter and Instrument Div., General Electric Co., Schenectady 5, N. Y.

Circle No. 36 on Reader Service Card

RUBBER PAD is claimed to eliminate up to 75 percent of vibration when used under machinery legs. Simple to use. Intended for small plant with few machines. B. F. Goodrich Co., 10 E. 40th St., New York 16, N. Y.

Circle No. 37 on Reader Service Card

MULTI-COAT FINISH is applied in a single coat. Any number of colors can be used. No primer necessary except when used on metal. United Lacquer Mfg. Corp., 1001 W. Elizabeth Ave., Linden, N. J.

Circle No. 38 on Reader Service Card

TAP CHANGING TRANSFORMERS are rated from 750 to 5,000 watts for furnace temperature control and general variable voltage use. Coarse and fine adjustment knobs. Hevi Duty Electric Co., Milwaukee 1, Wis.

Circle No. 39 on Reader Service Card

CONTINUED ON PAGE 198

DIE CASTING

is a moving picture of one economy after another

CASE HISTORIES FROM MT. VERNON FILES

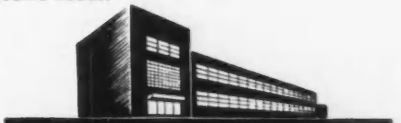
Run over any production picture that features die casting — and you'll find ECONOMY stars in every cast.

Here's a movie camera that testifies to that — and what's better than a movie camera for showing the evidence? Listen to it expose its views:

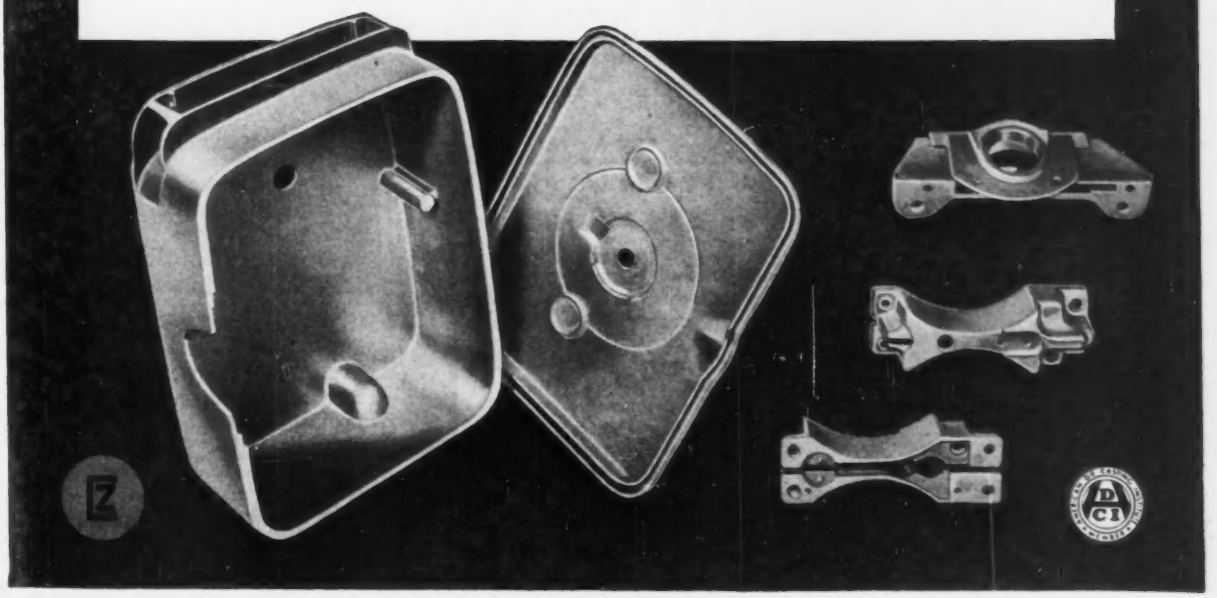
"Since I'm made by DeJur-Amsco Corporation to give perfect service anywhere on earth, and 'precision' is my middle name, every component part of me has to be strong, durable, and accurately produced. Not without due respect to economy, however... my field being a highly competitive one.

Because of this, my 2-piece housing, lens mount, and film gate block are all die cast. Die casting not only rapidly produces the intricate shapes, bosses, channels, slots and holes required, but furnishes these components so accurately that smooth operation and complete light-tightness are assured by precision fit of the parts. Even the critically uniform path required for the film as it passes through the gate, takes a minimum of machining to finish. Thus die casting saves time, material, money — while making possible a quality product."

Which gives good reason for focusing on die casting when next you consider a production procedure. And remember, we at Mount Vernon — who produced the parts mentioned above — have a wealth of accumulated experience and modern facilities... are always glad to make recommendations based upon your specific needs.



MT. VERNON
DIE CASTING CORP.
MT VERNON, NEW YORK



How to be SURE
When You Specify Springs
When we recommend a spring for a particular job you can be sure it's a sound recommendation. For many, many years we have specialized in designing springs and wire shapes for almost every conceivable type of application. Whether your requirements call for a standard design or a spring specially created for your product, you can count on us to provide the spring that is precisely right.
Would you like a copy of American-Fort Pitt's 28 page handbook of engineering data on springs? It's packed with facts and formulas —and it's yours for the asking.
AMERICAN-FORT PITT SPRING DIVISION
H. K. PORTER COMPANY, INC. McKees Rocks, Pa.
AMERICAN-FORT PITT
SPRINGS

Sewall GEARS
CUSTOM MADE
For EVERY NEED
For best results, get in touch with Sewall engineers while your product is still on the drawing board. Close tolerances are efficiently maintained in the production of racks... sprockets... spur... spiral... bevel... Zerol bevel... worm gears. We furnish prompt estimates on any quantity.
Send for STOCK SPUR CUT Catalog
E. B. SEWALL MANUFACTURING CO.
696 Glendale St. St. Paul 4, Minnesota.

Counter Records High Speed Shaft Revolutions

Speeds to 4,000 counts per minute are provided by a revolution counter. Built for industrial service and high speed operation, this new counter has an extra large shaft with oversize bearings, all steel housing with unbreakable plexiglass windows and large, light-weight number wheels, with sharp easy-to-read figures.

Standard models add one unit for each forward revolution of the shaft and subtract reverse revolutions. Also available to add ten units for one shaft revolution and other ratios.

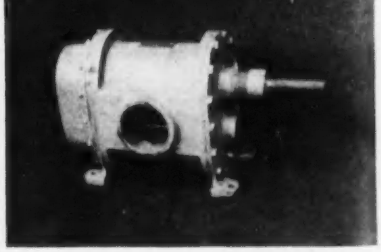
Available with top-coming or top-going shaft rotation; 5 or 7 number wheels; reset knob or key; right, left or double shaft extension. Overall dimensions, $4\frac{15}{16}$ in. long, $3\frac{15}{16}$ in. wide, and $2\frac{3}{8}$ in. deep.

Production Instrument Co., 702-10 W. Jackson Blvd., Chicago 6, Ill.

Circle No. 40 on Reader Service Card

Rubber Impellers Pump Difficult Liquids

A self-priming, positive displacement, rotary type pump is designed to handle a variety of difficult pumping applications. Impellers consist of two steel spiders on which an abrasive resistant layer of Neoprene, Hi-Car or Buna rubber (depending on service) has been bonded. Drive is furnished by steel cut machine gears keyed to the shaft and enclosed in an oil bath case which is isolated from the medium being pumped. The rubber affords the advantage of self priming suction lift of twenty feet and provides



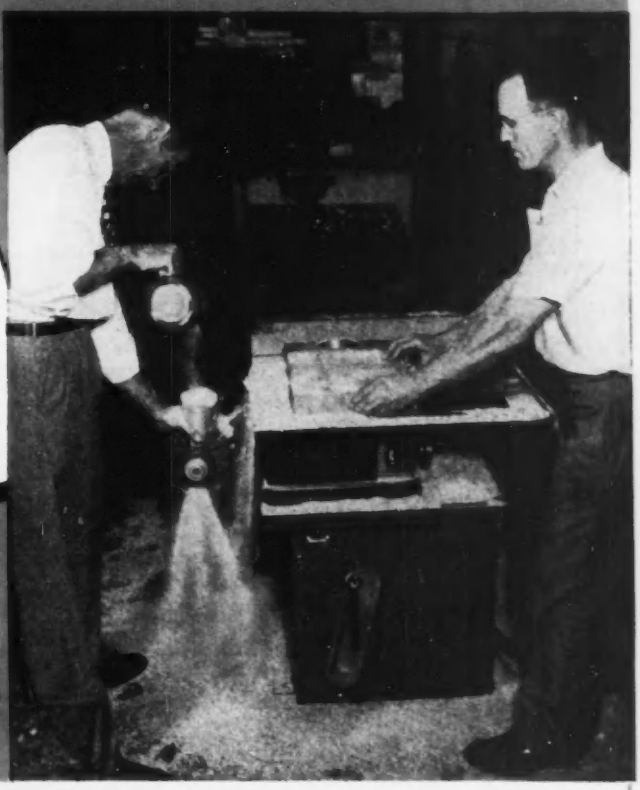
a resilient surface which permits the handling of viscous liquids, slurries and solutions containing mud, sand and abrasives without the usual wear accompanying such service. Since pumpage does not go thru the gears but is carried around the outer periphery of the case at slow velocity the erosive action of abrasive solutions is kept at a minimum.

Two models are in production, a

EMERSON-ELECTRIC Announces a Major Advance in Fractional Horsepower Motor Design!

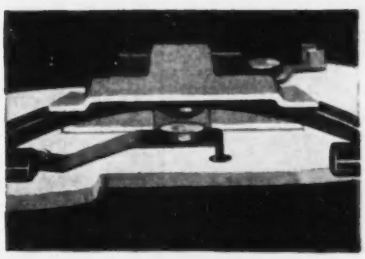
New Dustproof Centrifugal Starting Switch

PATENT APPLIED FOR

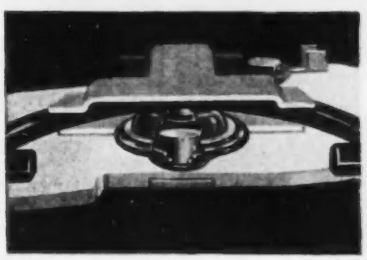


Tests show that even continuous pouring of sawdust directly into motor will not foul the improved Emerson-Electric starting switch! A conventional starting switch would fail to operate when contacts became clogged, and the motor would not start, causing possible serious damage to the windings.

Before:
The standard-type starting switch, mounted on the inside of the motor end shield, has exposed contact points, as shown here. Sawdust, dust and lint entering through the end-shield vents may lodge on the points and prevent contact . . . a possible cause of motor failure.



After:
Vital contact points are completely shielded. Insulating and metal washers are held firmly against the lower rim of the shield, by a spring arrangement. In this view the shield, washers and spring are cut away to show how they protect the contact points.



Switch contacts are completely protected . . . yet there is no change in other internal components, external motor design or dimensions.

Here's *news* that may eliminate the cause of motor failure for *your product!* This dustproof switch was designed for power-saw motors — which operate under the most severe dust conditions. It has proved a complete success. The device is simple and fool-proof, and it offers your product competitive advantages that you cannot obtain in any other motor.

This important development will give Emerson-Electric motors a marked superiority for many tool, appliance, and blower applications, using ventilated motors of split-phase or capacitor-start design. Your inquiry is invited.

THE EMERSON ELECTRIC MFG. CO.,
St. Louis 21, Mo.

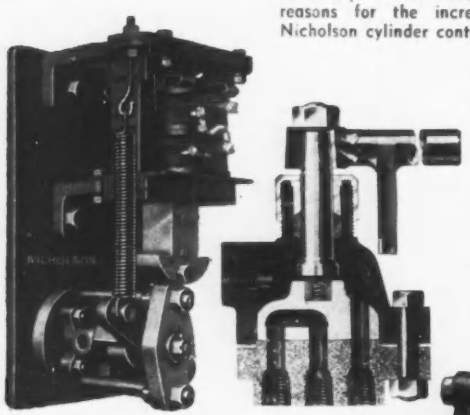
EMERSON ELECTRIC
MOTORS • FANS — APPLIANCES

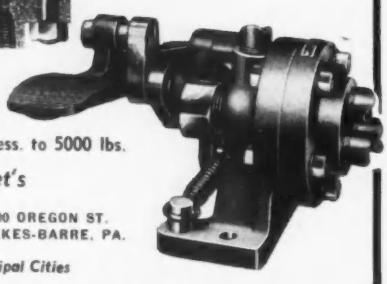
NICHOLSON CONTROL VALVES *feature*

EASY MAINTENANCE

A survey of maintenance men shows these 6 reasons for the increasing standardization on Nicholson cylinder control valves:

1) Specially treated hard seats; don't cut out.
2) Seating surfaces protected from dirt, grit.
3) Metals, gaskets and gland packing to suit service.
4) Generous stuffing box, easily repacked.
5) Heavy-bodied, take abuse.
6) Simplicity of design, minimizes maintenance time.





Nicholson standard and special valves in lever, foot, solenoid, motor types; for air, gas, oil, steam, water; size ¼" to 2½"; press. to 5000 lbs.

Bulletin 546 or see Sweet's

W. H. NICHOLSON & CO. 200 OREGON ST. WILKES-BARRE, PA.

Sales and Engineering Offices in All Principal Cities

"I'm Rugged! I'M MICRO-WEAVE TRACING CLOTH"

"Yes, sir, I can really take it! I can stand up under repeated erasures — hand or mechanical — and still take pencil smoothly *or* ink-over freely — *without* leaving 'ghost lines' — that's what I was made to do and that's *exactly what* I do — thanks to my finer, closer weave and unique surface."

MICRO-WEAVE Tracing Cloth exceeds all specifications in transparency — in minimum feathering — in maximum strength. Put it to any test on your drawing board and discover why it is first choice with so many draftsmen all over the country. Generous sample on request.

THE HOLLISTON MILLS, INC.
NORWOOD, MASS.
NEW YORK PHILADELPHIA CHICAGO

★ MICRO-WEAVE is backed by Holliston's 50 years of leadership and experience in developing special cloths for industry.

two inch with capacity of 100 gallons per minute and a three inch with capacity of 300 gallons per minute. Capacity is in direct ratio to speed so that flow may be varied. Heads are to 150 feet.

Lindquist-Reynolds Pump Co., 16263 Grand Ave., Bellflower, Calif.

Circle No. 41 on Reader Service Card

Tuner Provides Reception Of FM and TV Channels

Mechanically and electrically designed for replacement of switch-type TV tuners, the new component provides reception of FM as well as TV channels at a new low in cost. The input tuner employs a three-section spiral tuner plus antenna tuning. This combination provides excellent sensitivity and selectivity. The space required is identical with that of most leading switch-type tuners. Standard mounting holes further facilitate interchangeability. Electrically, it is designed to work into the if system of TV receivers using a separate sound if. It is available with variations in the mixer plate network, making it adaptable without alteration to the if system of other separate sound if television receivers.

The tuner provides particular emphasis on improved high band performance. Input impedance is 300 ohms. The inclusion of the sound trap with the tuner is optional as well as the choice of either 21.25 or 21.75 mes. sound center if. The tuner has low oscillator radiation and low noise figure. It comes complete with tubes (6BC5 rf tube and 6J6 mixer-oscillator), TV-FM dial scales and mixer plate network.

Electronic Parts Div., Allen B. DuMont Lab., Inc., 35 Market St., E. Paterson, N. J.

Circle No. 42 on Reader Service Card

Aluminum Rotor Cuts Motor Weight

Outstanding features claimed by the makers of a timing motor include a lightweight rotor with aluminum spider of large diameter which attains synchronization at one-twelfth revolution, or on the second or third alternating impulse, resulting in greater accuracy of performance.

All bearings are lubricated by a sealed-in supply of special oil, distributed by capillary circulation surrounding all moving parts. The patented two-piece field structure has wide and narrow poles, with rotor

operating between inner and outer field poles which are properly shaded with heavy, continuous copper rings for maximum starting and running torque.

Double bearings are used on all shafts, including the reduction train and rotor shaft. The motor may be mounted in any position and pulls up to 8 in-oz continuously at 1 rpm. It operates with equal efficiency in temperatures ranging from 40 below zero (F) to 140 above.

Motor is approved by the Underwriters' Laboratories, Inc. and by the Canadian Engineering Standards Assn. for 110 and 220 volts.

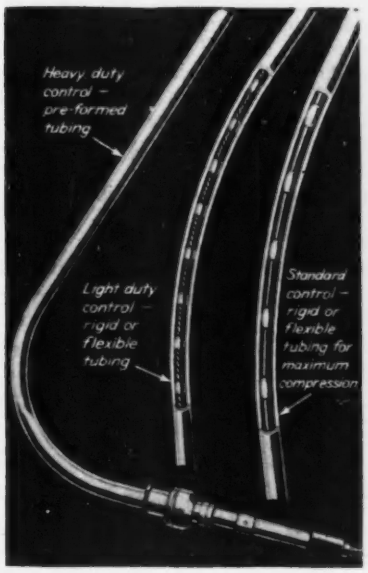
Hansen Mfg. Co., Princeton, Ind.

Circle No. 43 on Reader Service Card

Push-Pull Controls Have Variety of Applications

A line of pre-assembled and preformed push-pull controls, are improved models of the push-pull controls which have had successful wartime installations.

Basic elements of the controls are a


Heavy duty control — pre-formed tubing
Light duty control — rigid or flexible tubing
Standard control — rigid or flexible tubing for maximum compression

SPECIFY PARKER
TRIPLE-LOK
THE J.I.C.
TUBE FITTING

- 37° flare angle
- Dryseal pipe threads
- has actually passed rigid AN-F-47 performance test as required by J. I. C. standards. Certified laboratory test report available.

MADE BY THE WORLD'S LARGEST MANUFACTURER OF LEAKPROOF TUBE FITTINGS.

- precision-made to be *leakproof* under even the severest operating conditions.
- holds *leakproof* beyond tube bursting pressures.
- 3-piece design—remains *leakproof* after repeated reassembly.
- accommodates full range of standard tube wall thicknesses.
- made in steel, stainless steel, also aluminum alloy.
- shape bodies machined from forgings, other parts from bar stock.
- stocked by distributors across the nation.
- write for Parker "Industry Standard" Tube Fittings Catalog 203.

THE PARKER APPLIANCE COMPANY
17325 EUCLID AVENUE, CLEVELAND 12, OHIO
5827 W. CENTURY BLVD., LOS ANGELES 43, CAL.

Parker
TUBE FITTINGS
VALVES · O-RINGS

The PARKER Appliance Company
Cleveland 12, Ohio

Please send my free copy of Tube Fittings Catalog 203

Name _____

Company _____

Address _____

City _____ State _____

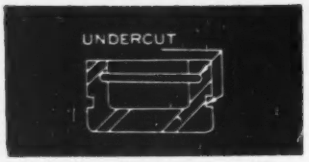
Materials Engineering Facts

HINTS FOR MOLDED PARTS OF ACE HARD RUBBER

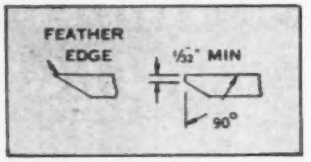
Ace Hard Rubber is a widely used thermosetting plastic, combining outstanding chemical, physical and electrical characteristics—including high tensile strength, low moisture absorption, high dielectric strength and good machining qualities.

You can choose from Ace compounds which offer tensile strength up to 9700 psi, dielectric strength to 613 v/mil, heat resistance to 300° F. water absorption as low as 0.04—with complete (among world's largest) Ace facilities for design, molding, extruding, machining, finishing, etc.—for parts made the most economical way to your exact specifications. Also Ace plastics such as Parian (polyethylene), Saran, Ace-Tex, etc.

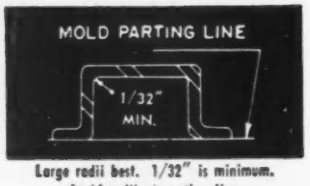
Always check your Ace Hard Rubber and Plastics Handbook when selecting materials. If you haven't a copy of this valuable 60-pg. manual, *write today —it's free.*



Undercuts usually cheaper to machine than to mold

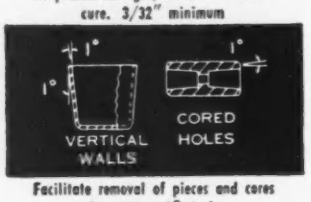


Avoid fragile edges, 1/32" is minimum

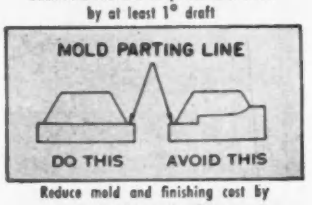


Large radii best. 1/32" is minimum. Avoid radii at parting line

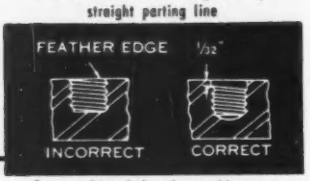
Keep sections light, uniform for short cure. 3/32" minimum



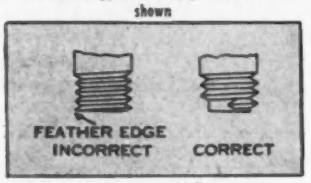
Facilitate removal of pieces and cores by at least 1° draft



Reduce mold and finishing cost by straight parting line



Prevent chipped threads; provide recess shown



Start male threads 1/32" from end

ACE AMERICAN HARD RUBBER COMPANY

HARD RUBBER and PLASTICS

Since 1851

11 MERCER STREET • NEW YORK 13, N. Y.

moving member, or linkage, enclosed in either a rigid or flexible casting, or a combination of both. Two types of linkage are available, $\frac{1}{4}$ in. OD light duty control for light aircraft and industrial application and a $\frac{5}{16}$ in. OD and a $\frac{7}{16}$ in. OD for combined accuracy and strength.

Standard end fittings available are sliding rods (threaded, plain and hooked end); clevis ends; quick-detach coupling units; ball and socket end fittings; radiant units (converting linear to rotary motion); friction locks (for locking control in any position of its stroke); adjustable friction heads (for varying control resistance to movement) and a vernier head for combining adjustable friction head and friction lock functions and providing vernier adjustment.

When tested under temperatures ranging from minus 72 deg F to 160 deg F, the controls were unaffected over tens of thousands of cycles. Controls are lubricated for approximately a million cycles.

Industrial Div., Simmonds Aerocessories, Inc., Tarrytown, N. Y.

Circle No. 44 on Reader Service Card

Timing Motor Fits Circular Mounting

This synchronous motor is perfectly round and will fit into a 2 in. circle mounting, making it simple to install in confined spaces. An input of 4 watts develops high starting and running torque.

The motor is fully synchronous for an extended range of voltage fluctuations. It is furnished as standard in a large variety of speeds and several types of output shafts. The gear train and rotor assembly section is separate from the coil and stator assembly.

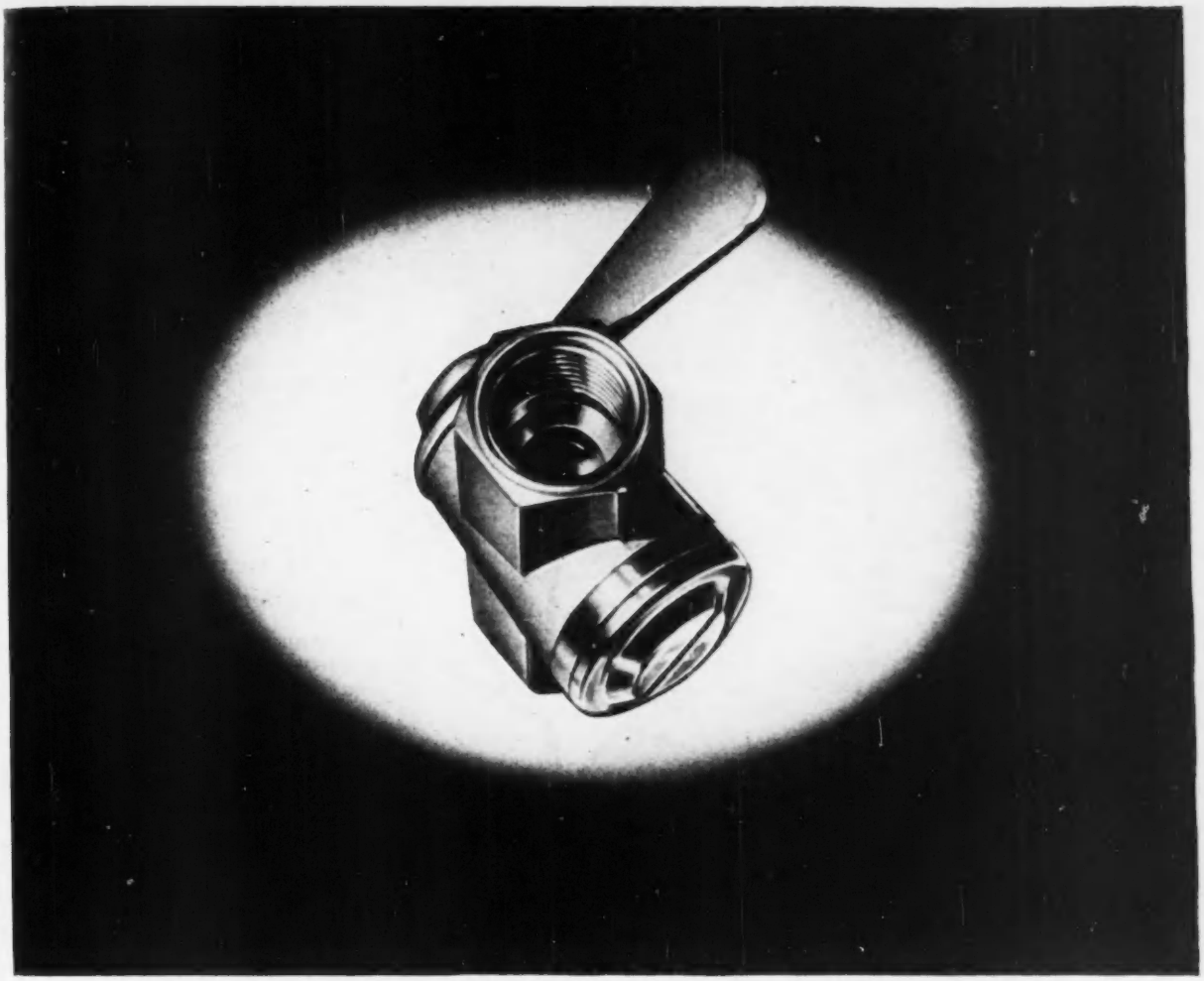
An efficient capillary oiling system is designed to insure long life and quiet operation. The motors are built to high standards, yet the design enables the manufacturer to produce these units to sell at a price schedule adaptable for use in timing, sequency and motion devices, where a sturdy, compact and small unit is desirable.

The Bristol Motor Co., Bristol Bldg., Old Saybrook, Conn.

Circle No. 45 on Reader Service Card

Toggle Switch Parts Are Entirely Enclosed

A compact toggle switch body measures 1 3/16 by 1 3/16 by 1 in. and working parts are entirely en-



KOHLER PLUG VALVE. *Forged Aluminum Body With Precision-Lapped Steel Plug.*

KOHLER PRECISION PARTS

Aircraft, Industrial, Automotive

KOHLER PRECISION PARTS are used by manufacturers of Diesel engines, automobiles, trucks, tractors, jet engines, military, commercial and private aircraft, and for agricultural and industrial test equipment.

With an organization experienced and skilled in precision workmanship, Kohler maintains full facilities for forging, casting, machining, finishing, assembling, and testing.

Kohler engineers will develop precision valves and fittings for volume production to meet special applications. Write for a catalog and send your specifications or requirements.

Kohler Co., Kohler, Wisconsin. Established 1873.

KOHLER OF KOHLER

PLUMBING FIXTURES • HEATING EQUIPMENT • ELECTRIC PLANTS • AIR-COOLED ENGINES

there's always room for one more!

★ Each of these products is a hit in its field because an imaginative manufacturer capitalized on the unique properties of VINYLITE Brand Plastic flexible molding and extrusion compounds.

The same profit possibilities are also available to you. Here's how to make the most of them.

Take a new look at your product—either present or contemplated. Can it use:

Flexibility? VINYLITE Plastic flexible compounds are available in varying degrees of flexibility from soft and rubbery to semi-rigid. They withstand long flexing without cracking.

Good looks? These materials are available in the widest possible range of colors.

Durability? They are non-fading . . . resist abrasion . . . give longer wear than any of the older elastomeric materials.

Resistance to oils, chemicals? VINYLITE Plastics are highly resistant to oils, greases, alkalies, most acids.

Low cost? These compounds are adaptable to high speed mass production either through injection molding or extrusion.

If your product can use any combination of these properties, it will pay you to investigate VINYLITE Plastic flexible molding and extrusion compounds.

Write for helpful 24-page illustrated booklet outlining the major applications, giving complete technical data. Ask for "VINYLITE Resins and Plastics—Extrusion and Molding Materials." Address Department JA-10.

Vinylite BRAND PLASTICS

BAKELITE DIVISION UCC

BAKELITE DIVISION
Union Carbide and Carbon Corporation • 30 East 42nd Street, New York 17, N. Y.

closed for protection and safety in operation. The switch snaps into a hole where spring clips hold it in place. The standard clips are designed to give a snug grip on panels ranging in thickness from 0.025 to 0.100 in. The shoulders provide an overlap to compensate for wide tolerances in finish dimensions of the installation holes.

The switch is rated at 15 and 20 amps, 125 v; 10 amps, 250 v, a-c, and up to ¾ hp at 250 v, a-c. The plastic body is standard in brown, black and white. Other colors are special.

Hart Mfg. Co., 110 Bartholomew Ave., Hartford, Conn.

Circle No. 46 on Reader Service Card

Test Set Measures Conversion Loss

A portable self-contained mixer crystal test set for measuring conversion loss and noise temperature of silicon crystals is being produced. The compact instrument, particularly suitable for production testing, incoming inspection, and field tests, has a correlation accuracy of plus or minus ½ db on conversion loss measurements and ½ on noise temperature mean deviation. The test set, available for immediate delivery, is intended for use at or below 10,000 mc for direct indication and above 10,000 mc for relative indication.

Airborne Instruments Laboratory, 160 Old Country Road, Mineola, N. Y.

Circle No. 47 on Reader Service Card

Plastic Pipe Takes Increased Pressure and Suction

Pipe is extruded from specially compounded polyethylene in heavy thicknesses for high pressure and low suction.

It is recommended for industrial solutions and corrosive gases and vapors. Air duct and electrical conduit applications are expected. The pipe is also used for suction pump sludge transfer and in mine water service. Its flexibility cuts the number of fittings required. Molded plastic fittings make metal-to-plastic or plastic-to-plastic connections.

Almost every commercial fluid can be handled with the exceptions of some petroleum spirits, concentrated alcoholic beverages, and the concentrated halogens.

Carter Products Corp., 10327 Meech Ave., Cleveland 5, Ohio

Circle No. 48 on Reader Service Card

OUR READERS SAY

Percent Instead of Degrees

To the Editor:

On page 151 of the September 1950 issue of PRODUCT ENGINEERING, you published an abstract from an earlier article on our development of the coal-fired gas turbine. The editor who wrote the abstract did a very competent job, but one error appears in the eighth paragraph.

In commenting on the speed of the turbine, the original article stated that the speed remained quite constant, although temperatures at the combuster outlet varied as much as 25 F from mean values. The variation is cyclical in nature and is of such rapidity that there is no visible effect upon turbine speed. However, if there were a variation of anything like 25 percent, we would certainly see it. Our variation is actually 25 F out of about 1,200 F or about 2 percent.

—JOHN I. YELLOTT
Bituminous Coal Research, Inc.

Ed—Apologies are in order for the error which the reader so clearly points out. The temperature variation is 25 F, not 25 percent.

Exterior Gold Finish

To the Editor:

Prior to the advent of such non-tarnishable finishes as stainless steel and chromium plate, the external polished metal fittings of our cars were made of solid bronze or nickel alloy. Frequently we are asked to reproduce the equivalent of the bronze finish in a non-tarnishable form.

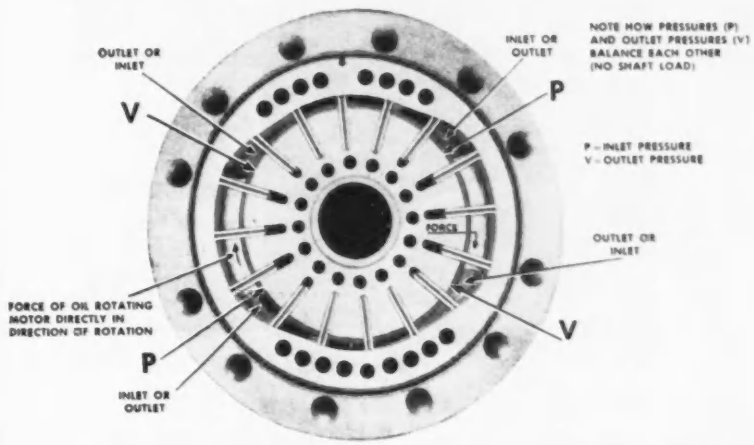
As a subscriber to your journal, we take the liberty of asking whether you could suggest a source to which we should direct our inquiries for a non-tarnishable gold colored metal or for a means of providing a durable protective coating or metal of the desired color that is not tarnishable when exposed to out-of-doors weather conditions.

In Switzerland, there is a metal which is used in the jewelry trade for fountain pens and self-propelling pencils. This metal appears to be non-tarnishable even when unlacquered. We have no experience of its behavior

Dudco VANE-TYPE MOTORS

Will save YOU...Money, Space, Time!

Dudco Vane-Type Motors operate by direct force of the fluid against the vanes. The fluid enters through ports which are diametrically opposed, to balance shaft loads, and discharges in the same manner. Notes included with the figure below relate to fluid motor construction. Pump operation is similar, with the dual vanes bearing radially outward with positive forces which, while adequate for pressure sealing, are balanced hydraulically to eliminate undesirable wear producing thrust against the ring.



CHECK THESE FEATURES

- **DUDCO MOTORS operate down as low as 10 or 15 RPM at high torques often eliminating the necessity for gear reduction units**
- **Starting torque is over 93% of theoretical torque**
- **Torque ratings from 9 lb.-in. to 550 lb.-in. per 100 PSI available**
- **Quiet operating even at 3600 RPM**
- **Smaller size for ease of application**
- **Overall mechanical efficiency 85% at 1200 RPM, 77% at 100 RPM after long use**
- **120 HP available at 800 RPM**
- **Hydraulically balanced for long life—without any mechanical means for holding vanes out against the stator ring**
- **Rugged reversible service is common application**

CHECK THESE USES

- **Lumber (Resaws, slitters, conveyors, carriage feed works).**
- **Cannery (Box dumpers, fruit conveyors).**
- **Machine Tools Thread tapping, lapping, indexing, bending machines, etc.).**
- **Chemical (Centrifuge, mixers, agitators, screw conveyors).**
- **Mining Equipment (Traction, Shoveling).**
- **Trucks (Hose reeling, conveyors).**
- **Railroad Work Equipment.**
- **Drilling Rigs (Rotary table, cathead winch)**
- **And many more**

DUDCO'S over 100 field representatives stand ready to assist in the application of Vane-Type Motors and Pumps to your equipment.

Write today for our complete catalog and our handy fluid motor data sheets showing HP output and speed vs. volume and pressure required.

Dudco PRODUCTS COMPANY

1784 E. Nine Mile Road ★ Hazel Park, Mich.

Please send me catalog and motor data sheets:

Name ..... **Title** .....

Company .....

Address .....

City ..... **Zone** ..... **State** .....

Another Contribution to Industry through Powder Metallurgy by BOUND BROOK

IF YOU USE *TINY* PRECISION BEARINGS IN *LARGE* QUANTITIES...TRY "COMPO" SELF-LUBRICATING BEARINGS FOR *CLOSE TOLERANCES AT LOW COST!*

"COMPO" For products such as electric shavers, clocks, meters and many types of instruments, Self-Lubricating Bearings provide these advantages:

- extremely low unit cost
- quiet operation
- lubrication for long, dependable service
- simplified assembly

BOUND BROOK "COMPO" bearings and structural parts are:

- made from pure metal powders
- die-formed to shape
- alloyed at high temperatures
- finished to precise dimensions, AND
- vacuum-impregnated with lubricant

CONSULT OUR ENGINEERS

BOUND BROOK "COMPO" • "POWDIRON" • BOUND BROOK

when exposed to the external atmosphere. This is the sort of metal we have in mind.—H. I. F. EVERNDEN
Rolls-Royce Ltd.
Derbyshire, England

Ed—One possibility is a special bronze alloy that is supposedly non-tarnishable. Details concerning this material can be had from any one of the brass and bronze institutes. However, Mr. Evernden may be thinking of a die cast part or product, and this bronze alloy is definitely not one of the die cast materials. It must be machined to final shape.

Another possibility is a thick gold plate over a properly prepared zinc die casting. Gold plate, provided that it is thick enough, is practically non-tarnishable and, on a product like a Rolls-Royce, would not be too costly.

The Chrysler Corporation has made several medallions for interior trim, using a thin gold plate over a die casting. Mr. J. C. Fox, Chief Metallurgist, Doehler Jarvis Corporation, 386 Fourth Avenue, New York City can supply specific details on the preparation of the zinc base casting and probably could suggest the minimum thickness of the gold plate.

Multi-Stylus Recorder

To the Editor:

We noticed in the July 1950 issue of PRODUCT ENGINEERING that Mr. M. T. Nance of Shell Oil Company has requested information concerning a multi-stylus recorder. We also note you have given information to him concerning various companies who can supply this type of equipment.

We would like to apprize you of the recorder which we have developed. Our company is producing equipment which is capable of remote indication of variables such as pressure transmitters, accelerometers, and temperature probes. Our recorder has a direct-writing characteristic made possible by the use of heated stylii operating on heat-sensitive paper. The trace is characterized by a very fine line, thereby providing increased accuracy in recording.—H. D. WRIGHT
G. M. Giannini & Co., Inc.

High-Pressure Pump

To the Editor:

We are in need of a small high pressure air pump that will deliver approximately two cubic inches of air per minute at 3,000 psi. A booster that could be utilized with a 1,500-psi

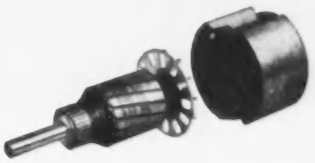
Design for Compactness, Low Weight... one of 6 Advantages of...

Lamb Electric

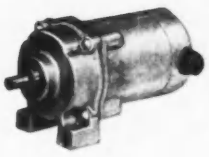
special application fractional horsepower **MOTORS**

A motor engineered for compact assemblies . . . as computing machines and other types of motor driven office equipment.

Face mounted, pinion shaft motor for applications where high torque is required at high speed.



Universal motor parts for portable electric tools, garage equipment and other industrial devices.



Pocket-size gear motor for machine applications requiring low output shaft speeds with relatively high torque.

Our 35 years' experience in designing motors for minimum weight and space factor frequently enables us to reduce *product* weight, yet maintain optimum performance, and lower cost.

These are among the advantages of using Lamb Electric Motors. Designed to meet the exact requirements of your application, they provide:

IN THE MOTOR...

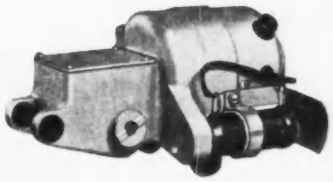
1. Reduced cost, weight, space.
2. Exact mechanical and electrical characteristics.
3. Thorough dependability.

IN THE PRODUCT...

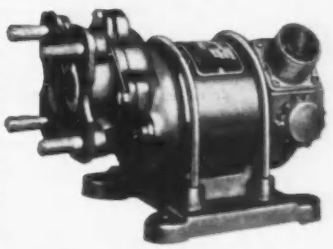
4. Better performance.
5. Improved eye-appeal.
6. Compactness, less weight.

Our engineering department will be glad to work with yours in obtaining these benefits.

Intermittent high torque motor with low weight factor; for aircraft and many other applications.



Series universal motor with inbuilt gear reduction and drive shaft at right angles to armature shaft.



Compact and dependable explosion-proof aircraft geared fuel transfer pump motor. Adaptable to many other applications.

THEY'RE POWERING AMERICA'S *Finest* PRODUCTS

THE LAMB ELECTRIC COMPANY, KENT, OHIO

Lamb Electric

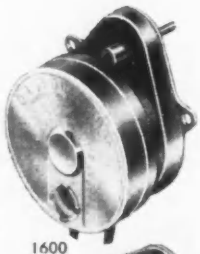
SPECIAL APPLICATION FRACTIONAL HORSEPOWER **MOTORS**

Timing Ideas

WHEN TIMING COUNTS

You can always count on Haydon to provide the right motor for your timing job. Timing Headquarters offers a broad line of efficient, economical timing motors and timers produced by specialists who have no other interest than to provide the best in timing.

LOW COST VERSATILITY—1600 SERIES: Specifically designed as a standard component for the widest possible range of timing applications with 79 speeds from 300 rpm to 1 revolution per 4 hours.



1600

COMPACTNESS AND SLOW SPEED—4400 SERIES: Designed for small size and low cost in applications requiring slower speeds from 6 hours to 7 days per revolution.

4400

HEAVY DUTY DEPENDABILITY—3100 SERIES: For control and instrument applications that require a heavy duty train and speeds from 1 hour to 14 days per revolution.

SUPERIORITY FEATURES

Slow (450 rpm) rotor speed makes for quiet operation and long life. Unusually small. All motors totally enclosed. Separate rotor and reduction gearing lubricating systems permit selection of best methods and lubricants, control circulation, insure against leakage. Operates continuously in any position. Simple to mount, entire face of motor can be supported securely against mounting surface. Standard, interchangeable design in only 2 motor series with speed range from 300 rpm to 1 revolution in 7 days.

3100

For complete design and engineering specifications, write for catalog: Timing Motors No. 322 — Timers No. 323 — Clock Movements No. 324. Yours without obligation.

HAYDON AT TORRINGTON

HEADQUARTERS FOR TIMING

HAYDON Manufacturing Co., Inc.

3135 ELM STREET

TORRINGTON, CONNECTICUT

SUBSIDIARY OF **GENERAL TIME** CORPORATION

pump to deliver the required pressure can also be used. Do you have any information as to where we might obtain the required pump or booster.

I have been told that PRODUCT ENGINEERING carried an article a short time ago on a pneumatic system that operates at 3,000 psi. If there was such an article, can you tell me the particular issue that contained the article?—R. GAUMER

U. S. Naval Ordnance Test Station

Ed—It is believed that the first article that has been published discussing pneumatic systems which go to these pressures was "Choice of Operating Pressures for Pneumatic Systems" by H. F. Gerwig, Pneumatic Design Engineer of Convair. Part I of this article appeared on page 127 of the August 1949 issue of PRODUCT ENGINEERING and Part II on page 139 of the September 1949 issue. Another article on this subject is in this issue of PRODUCT ENGINEERING on page 91. At present in the "Convair-liner", there are in use two 3 stage, piston type, engine driven compressors with a capacity of 5 cfm at 1,500 psi outlet pressure. A 3,000 psi compressor is currently in development. By getting in touch with these gentlemen you may be able to get further information as to the stage of development of this latter unit.

The Air Reduction Company in New York may be able to give you some help. If it proves necessary to go to a booster to get the required pressure, it is probable that either Charles Elmes Company in Chicago or the Watson-Stillman Company of Roselle, New Jersey would be able to design such a unit.

Testing Truck Wheels

To the Editor:

On page 118 in the August 1949 issue of the American journal, PRODUCT ENGINEERING, there appeared an article on technical progress in the rubber industry which reviewed a machine developed for testing industrial truck wheels. We shall be glad to know whether it is possible for you to provide us with such further details as the name of the firm supplying this machine or the name of a firm which could carry out the necessary tests.

—DEXINE RUBBER & ELBONITE LTD.
Lancaster, England

Ed—The article, "Friction Characteristics of Industrial Truck Wheels", describes in some detail the way in which the testing machines are made. These

THE *Magic Touch*

. . . that Increases Sales with MADISON-KIPP *Zinc and Aluminum* DIE CASTINGS

RED DEVIL NO. 30 PAINT CONDITIONER WITH SPRING MOUNTED COUNTER BASE. MANUFACTURED BY RED DEVIL TOOLS, IRVINGTON 11, NEW JERSEY, U. S. A.

THE legions of amateur painters have learned from experience to buy paint at stores equipped with mechanical mixers.

The paint store managers have learned from experience to buy Red Devil Paint Conditioners to do the essential job of paint mixing for their customers.

The Red Devil engineers know the great value of adding eye appeal, strength, lightness and quality to their products with Madison-Kipp zinc and aluminum die castings.

If you use metal parts in production quantities, you may find it profitable to send data on to us at our home office in Madison.

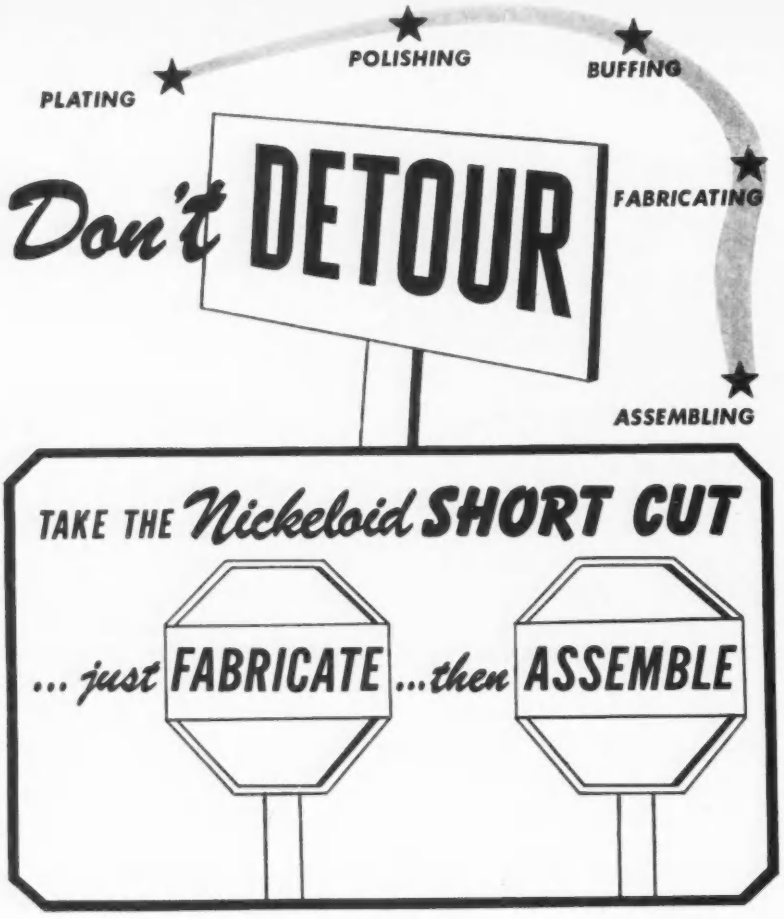
MADISON-KIPP CORPORATION

206 WAUBESA STREET, MADISON 10, WIS., U.S.A.

ANCIENS ATELIERS GASQUY, 31 Rue du Marais, Brussels, Belgium, sole agents for Belgium, Holland, France, and Switzerland.

WM. COULTHARD & CO. Ltd., Carlisle, England, sole agents for England, most European countries, India, Australia, and New Zealand.

- *Skilled in* DIE CASTING *Mechanics*
- *Experienced in* LUBRICATION *Engineering*
- *Originators of Really High Speed* AIR TOOLS



a finer **MATERIAL** *... a faster* **METHOD**

NICKELOID Pre-finished METALS provide the shortest distance between raw material and finished product. You simply fabricate, then assemble. Product beauty goes up, production costs go down. Choose from sheets or coils in a wide range of gauges, and tempers, plated one or two sides, bright or satin finishes of Chromium, Nickel, Brass or Copper, plated to base metals such as Steel, Zinc, Brass, Copper or Aluminum.

Write for new "Flow-Chart" showing how Nickeloid Pre-Finished Metals eliminate costly production operations.

NICKELOID METALS (Pre-Plated)
NICKEL · CHROMIUM · BRASS · COPPER
Electro-plated TO ALL COMMON Base METALS
ALSO LACQUERED COLORS
SHEETS AND COILS
OLDEST AND LARGEST PRODUCERS OF PRE-PLATED METALS

Established 1898
AMERICAN NICKELOID COMPANY
PERU 4, ILLINOIS

machines are not built commercially but were developed by the American Pulley Company for use in their own operations. It would be relatively simple to duplicate such machines.

An Old Chestnut

To the Editor:

In your February 1934 issue, you published an article by C. H. Grill, on the design of worm gears which I have found very interesting and useful.

It appears that formula (1) on page 63 is in error as to the projected tooth area and that the worm addendum should be doubled. The working depth of the tooth times its arc length would then be closer to the actual projected area.

—CHAS. F. VAN HOOK
The Watson Machine Company

Ed—The formula is not in error, but perhaps the author should have presented it more clearly. As the reader correctly states, the projected tooth area is equal to the arc length of the tooth along the pitch circle circumference times twice the worm addendum. This can be set up in the terms of the formula given in the article as:

$TA = Cw \times 2a \times \alpha/360$

TA = projected tooth area, sq in.
α = angle subtended on the worm by the gear, deg
Cw = worm pitch circle circumference, in.
a = worm addendum, in.

This then reduces to the formula as given in the article:

$$TA = Cw \times a \times \alpha/180$$

Circuit Breakers

To the Editor:

I am seeking information concerning deion air circuit breakers. I know that the American firms, General Electric, Westinghouse Electric, and Delta-Star Electric are manufacturing these breakers, and I should appreciate receiving catalogs from these firms concerning the matter mentioned above.

—KAARE REINHOLDT
Drammen, Norway

Ed—Letters have been sent to these American firms requesting that this information be forwarded to the reader.

Tube Fittings Data

To the Editor:

Has the McGraw-Hill Publishing Company printed a book or pamphlet

CLARE RELAYS

Here Is What CLARE Hermetic Sealing Means:

After assembly in the container, the enclosure is attached to a high vacuum pump and pumped down to a few microns pressure to remove all traces of moisture and gases.

While under this extreme vacuum, the enclosure and seals are tested for leaks by means of a Mass Spectrometer—a device so sensitive that it can detect a leak so tiny that more than thirty-one years would be required for one cubic centimeter of air to pass through it. This highly refined method of leak testing causes rejection of many enclosures which could pass the usual immersion tests without detection.

For most applications, the enclosure is then filled with dry nitrogen, which has a relatively high arcing potential.

Write for CLARE Bulletin No. 114

CLARE Hermetically Sealed Relays Protect Against These Conditions:

- Moisture, High Humidity and Ice
- Salt Air and Spray
- Fungus Growth
- Varying Air Pressure
- Variation of Air Density
- Dust and Dirt
- Corrosive Fumes
- Explosive Atmospheres
- Tampering

Clare Hermetically Sealed Relays are *air-tight* so that no gas or spirit can enter or escape.

This ideal condition, now available to every user of CLARE hermetically sealed relays, is the result of many years of painstaking research by the CLARE organization to produce a perfectly sealed relay at a reasonable cost to industrial relay buyers.

Hermetically sealed in an ideal atmosphere of dry inert gas, they are permanently immune to the difficult climatic and environmental conditions responsible for 95% of the failures of exposed electrical apparatus.

CLARE has today—or can provide you with—the hermetically sealed relay that you require. Over forty different series of CLARE hermetically sealed relays are described in Bulletin No. 114. Within each series, innumerable variations of coil and contact specifications are possible. Numerous other special sealed-relay units are also available.

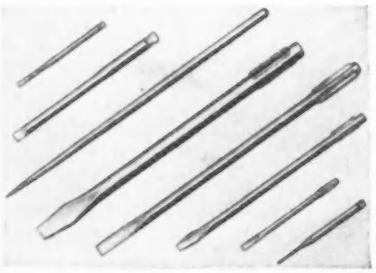
Clare sales engineers are located in principal cities to assist you in the selection of just the right relay for your specific requirement. Look them up in your telephone directory or write: C. P. Clare & Co., 4719 West Sunnyside Ave., Chicago 30, Illinois. In Canada: Canadian Line Materials Ltd., Toronto 13. Cable Address: CLARELAY.

CLARE RELAYS

PROMPT DELIVERY

on precision metal parts made by

TORRINGTON



You save time and money when you rely on our high productive capacity and special equipment to turn out precision parts made to your order.

We produce thousands of different parts. For example, screw driver and ice pick blades are made to order for assembly in any type of handle. Driver points keystone or cabinet. Uniformly ground edges. Winged, fluted, formed or milled shanks. Plain, blued or nickel plate finish.

We can make for you many such parts as instrument shafts and pivots, special rollers, studs, dowel pins, surgical and dental instruments, pen and pencil barrels, special needles for textile, leather, paper machines, knurled mandrels and spindles, etc.

Send your prints and specifications today for a prompt quotation.

THE TORRINGTON COMPANY
Specialty Department
554 Field Street • Torrington, Conn.

Makers of
TORRINGTON NEEDLE BEARINGS

that contains all the dimensions, threads and angles for sweat, flared and compression tubing fittings that are in common use?

If McGraw-Hill does, I wish you would quote the name; if they do not, I would appreciate your sending me the name of some other source for such a book.—MORTON STIVISON
E. Hempstead, L. I., N. Y.

Ed—The McGraw-Hill Book Campany does not publish these data. However, The Formed Steel Institute, 1621 Euclid Avenue, Cleveland, Ohio, has in progress a booklet giving at least part of such information. Mr. H. Myles Brown, of that organization is the person to write.

The American Standards Association, New York City, has recently come out with Standard B16.18.1950, which deals with cast brass solder-joint fittings.

Synthetic Hydraulic Fluid

To the Editor:

We are interested in the article entitled "Synthetic Hydraulic Fluids" written by C. M. Murphy and W. A. Zisman which appeared in the September issue of PRODUCT ENGINEERING. We would like to obtain further information regarding the fluid, tricresyl phosphate, mentioned on page 113. Can you send us the names of manufacturers or sources of supply of this material.—A. B. RIDDIFORD, JR.
John S. Barnes Corp.

Ed—It is suggested that you get in touch with the authors of this article for further information concerning tricresyl phosphate. The authors can be reached at the Naval Research Laboratory, Washington 20, D. C.

Where Can I Get It?

A Potential Customer

To the Editor:

I am a subscriber to PRODUCT ENGINEERING and beg to ask your help. Because of dollar difficulties in Peru, our Board of Directors has decided to undertake the manufacture of conduit piping, both light and heavy, ½, ⅝ and ¾ in. diameter sizes by the cold rolling method in our plant.

Our difficulty is the welding of the seam. None of the companies here who represent U. S. welding equipment manufacturers can give me def-

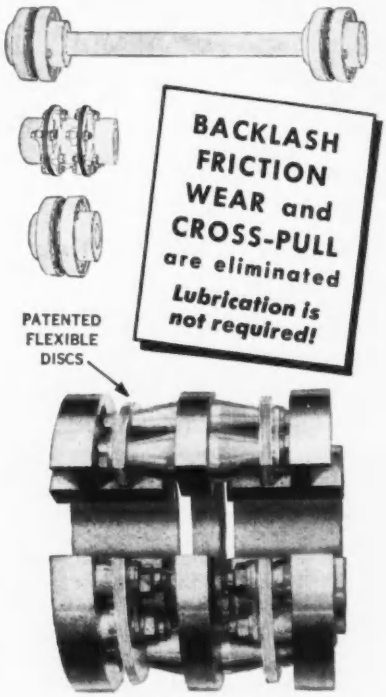
THOMAS *Flexible* ALL METAL COUPLINGS

FOR POWER TRANSMISSION REQUIRE NO MAINTENANCE

Patented Flexible Disc Rings of special steel transmit the power and provide for misalignment and end float.

Thomas Couplings have a wide range of speeds, horsepower and shaft sizes:
½ to 40,000 HP
1 to 30,000 RPM

Specialists on Couplings for more than 30 years



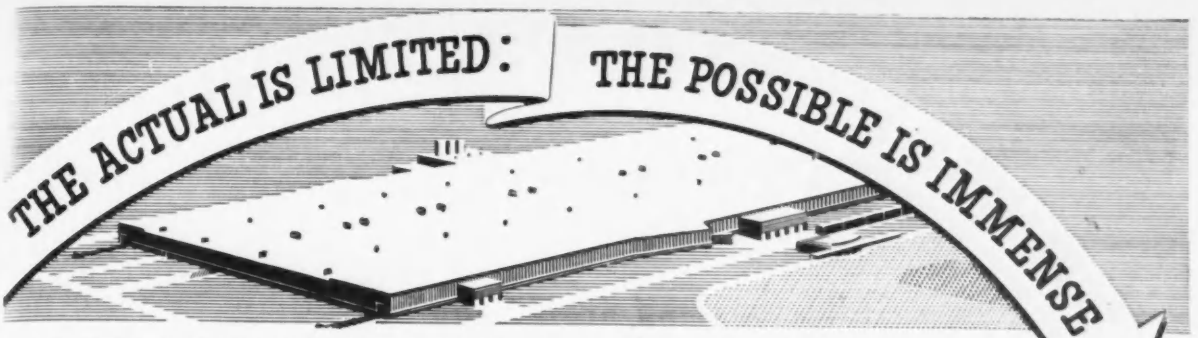
THE THOMAS PRINCIPLE GUARANTEES PERFECT BALANCE UNDER ALL CONDITIONS OF MISALIGNMENT.

NO MAINTENANCE PROBLEMS.

ALL PARTS ARE SOLIDLY BOLTED TOGETHER.

Write for the latest reprint of our Engineering Catalog.

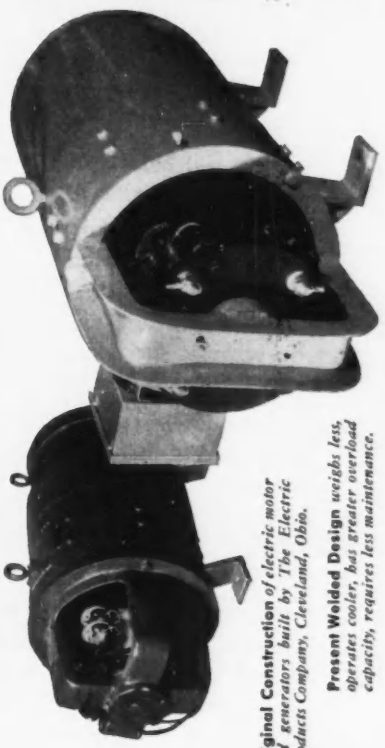
THOMAS FLEXIBLE COUPLING CO.
WARREN, PENNSYLVANIA



NEW LINCOLN PLANT CREATED BY INCENTIVE-INSPIRED CO-ACTION IN DEVELOPING POSSIBILITIES IN PRODUCT

WELDED DESIGN CUTS PRODUCTION COSTS
. . . opens new markets for machinery builder

Automatic Welding *of typical motor end bracket . . . produces smooth, uniform bead, saves manual cleaning operations to cut production costs.*



Original Construction *of electric motor and generators built by The Electric Products Company, Cleveland, Ohio.*

Present Welded Design *weighs less, operates cooler, has greater overload capacity, requires less maintenance.*

THE need for less material with welded design . . . fewer man-hours for machining and assembly is cutting production costs from 10% up to nearly 50% as on the part shown below.

The immensity of the possible is being indicated also in expanding opportunities for new business now being realized at The Electric Products Company, Cleveland, Ohio. By making possible product modifications at low cost to meet specialized needs, arc welding is daily tapping additional sources for sales.

Improved performance, cooler operation and less maintenance are resulting from better electrical characteristics of welded steel construction . . . from compact, rugged design permitting larger core openings and more efficient ventilation.

the ACTUAL

ORIGINAL DESIGN

increasing the YIELD

PRESENT WELDED DESIGN

the IMMENSITY of the POSSIBLE turns out to be a 50% SAVING IN COST

SEE HOW ARC WELDING IMPROVES PRODUCT ACCEPTANCE

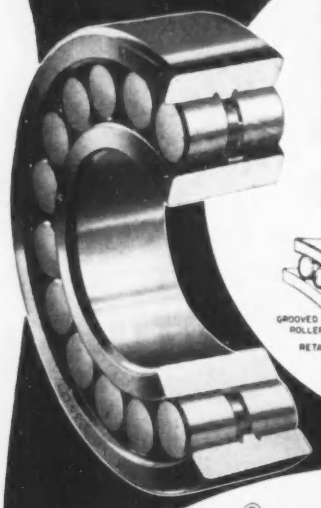
Machine Design Sheets free on request. Write Dept. 34,

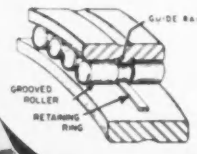
THE LINCOLN ELECTRIC COMPANY

CLEVELAND 1, OHIO

another FIRST in McGILL precision bearings

a NEW PRINCIPLE in roller bearing design

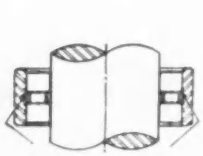




Rollers cannot skew. A guide rail integral with the outer race maintains alignment of the grooved rollers.

The NEW McGILL® GUIDEROL roller bearing

The Bearing with the Self-Aligning Rollers — a simplified roller bearing construction that automatically corrects roller skewing — without space wasting cages! Perfect roller alignment with resulting superior performance is maintained by this new principle of guided rollers. Even with reduced internal radial looseness a GUIDEROL bearing overcomes any tendency to bind under limited misalignment. Produced in needle and cylindrical type roller bearings. Write for information packed Bulletin today.



UNIVERSAL MOUNTING

Universal Mounting — from horizontal to vertical is possible. No cocking of rollers on vertical shafts because the grooved rollers are located by guide rail.

OPEN ENDS — SELF CLEANING

Open Ends — Self cleaning in applications permitting free circulation of oil or grease. Open faces set up flushing action inside the bearing.

GREATER ANGULAR RIGIDITY

Greater Angular Rigidity is provided by longer rollers that permit race and roller contact virtually to the outside edges of the races.

Write for your copy of the new GUIDEROL Bearing Bulletin No. GR-50. Address: McGill Manufacturing Co., Inc., Bearing Division, 201 N. Lafayette Street, Valparaiso, Indiana.

McGILL® precision bearings

inite information as to the type of a suitable welding machine which would be, I assume, of the roller electrode type. I should therefore be very grateful if you would put me in touch with manufacturers who could supply a suitable machine.

The production of our cold rolling machine is 2,500 metres per eight-hour day, and the welding machine would have to keep pace with this production.—L. W. B. BUDD
Talleres de Mecanica Industrial S. A. Lima, Peru

Ed—The Yoder Company, Cleveland 2, Ohio, should have equipment for such welding operations.

New Use For Adhesive

To the Editor:

In your November 1949 issue, page 5, reference was made to a waterproof adhesive that does not require heat or pressure and is suitable for cementing rubber, metal, cork, concrete, glass, wood, plastics, tile and asbestos. On the basis of this information, it appears that this adhesive would be satisfactory for use in sticking starch-filled and plastic-filled textile material to wood or cardboard mountings. We have done considerable experimental work on this particular application with other types of adhesives but have not obtained the degree of perfection desired.

We hope that you may be able to supply literature and information on this material.—WILLIAM H. BRITT
The Columbia Mills Inc.

Ed—Full information can be obtained from Adhesive Products Corporation, 1660 Boone Avenue, New York 60, New York, who manufacture this waterproof adhesive.

Strain Gage Specifications

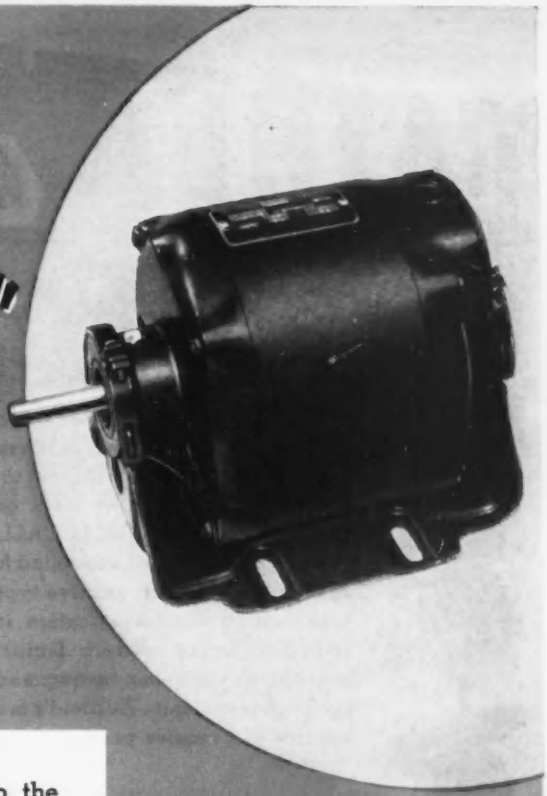
To the Editor:

We are extremely interested in the article appearing on page 113 of your July issue, entitled, "Sensing Small Motions . . . . . " by T. R. Breunich, Project Engineer, Sonntag Scientific Company.

This article attracted our attention because we are looking for a manufacturer who can furnish us with specifications on an electric strain gage that can be used to determine stress in steel members. We are particularly interested in exploring the practical aspects of using such a strain gage on a Tournahauler gooseneck, as manu-

On spec sheets everywhere

"DELCO PREFERRED"



...pays tribute to Delco quality and flexible supply

To win the approval of manufacturers to the point where their specification sheets indicate a preference for your product implies more than mere excellence of design. Delco, for example, has won the approval of appliance manufacturers in two ways:

First, by making millions of motors which have proved their dependability on millions of appliances over a long period of years.

Second, by maintaining a flexibility of supply which functions even when unusual conditions in the market necessitate changes in specifications and deliveries.

The experience of others is often the best guide. Delco motors, designed to meet specific torque and service requirements, are available in sizes from 1/8 h.p. up.

For complete information on all types and sizes of Delco motors contact the nearest Delco office listed below.

DELCO DAYTON OHIO

THERE IS A DELCO MOTOR FOR:

- Refrigerators
- Washing machines
- Dryers
- Furnace blowers
- Oil burners
- Stokers
- Water pumps
- Gasoline pumps
- Compressors
- Home work shops
- Fans (ventilating)
- Dishwashers
- Ironers
- Office machines
- Food grinders
- Separators

DELCO MOTORS

DELCO PRODUCTS

Division of General Motors Corporation, Dayton, Ohio

SALES OFFICES: CHICAGO • CINCINNATI • CLEVELAND • DETROIT • HARTFORD

FAIRFIELD

for FINE GEARS

. . . to meet your most exacting specifications

IF you use gears in the product you make, we believe it will pay you, as it has others, to become acquainted with FAIRFIELD — the place where fine gears are produced to meet your specifications EFFICIENTLY, ECONOMICALLY! Fairfield's production facilities are unexcelled for making all kinds of high precision, automotive type gears such as are now finding wide application in all branches of industry. Every modern facility is used in the processing, checking, testing, and inspection of the product. Investigate Fairfield's facilities today! Your inquiry will receive prompt attention.

Ask for this BULLETIN

Send for a copy of this interesting illustrated brochure describing Fairfield's gear making facilities.

Fine Gears
Made to Order:

- SPIRAL BEVEL
- STRAIGHT BEVEL
- HYPOID
- HELICAL
- HERRINGBONE
- SPUR
- WORM AND WORM GEARS
- DIFFERENTIALS

Here are a few of the industries for which Fairfield supplies fine gears *made to order*

Automotive • Construction Machinery • Agricultural Machinery • Aviation Equipment • Mining Machinery • Machine Tools • Oil Field Machinery Marine Equipment • Pumps and Winches Transportation

FAIRFIELD MANUFACTURING CO.

313 SO. EARL AVENUE LAFAYETTE, INDIANA

factured by the Le Tournea Company. The Tournahauler has 12 and 24 v direct current as well as 120 cycle, 240 v alternating current power available. We are also interested in this type of instrument as it may relate to tests on mill shafting.—NORMAN A. BLAICH
Grace Brothers, Ltd.
Honolulu, Hawaii

Ed—There are many concerns in this country who manufacture accessories for electric strain gage work, but the one company that can best help, not only with advice, but as to recommendations on instrumentation, is the Baldwin-Southwark Division of the Baldwin Locomotive Company, Philadelphia, Pa. Mr. Frank Tatnall, Manager, Baldwin Testing Division, can supply the answer to almost any problem dealing with strain gages.

Patent Complications

To the Editor:

Please mail us at the earliest time a copy of the No. 8-19, 1948 issue of PRODUCT ENGINEERING. Appearing in this issue is an article by S. W. Curtis entitled "Designing for Silver Brazing by Induction Heating."

We need this issue for a customer in connection with a matter of patents, and, therefore, it is important to know whether the article appeared before November 11, 1948.

—AUGUST LACHNER
Munchen, Germany

Ed—The article appeared in the August 1948 issue of PRODUCT ENGINEERING.

Insulated Screw Fasteners

To the Editor:

We have read with interest your article in the September 1950 issue of PRODUCT ENGINEERING entitled "Insulated Screw Fasteners for Electrical Equipment" by B. G. Forman, Research Director, Forman Insulating Screw Corporation.

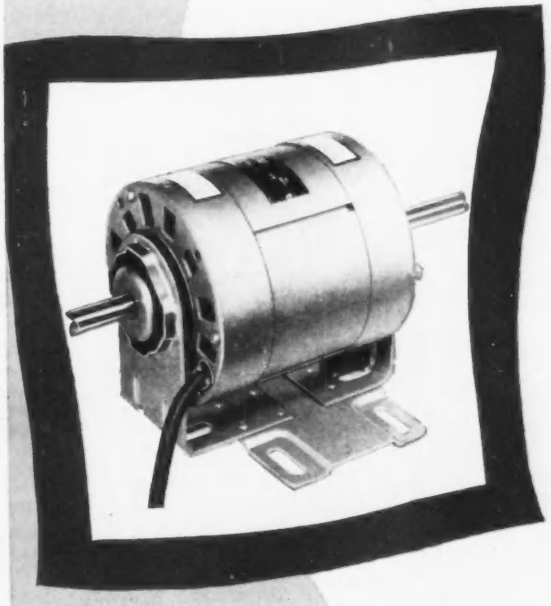
As we would like to obtain further information regarding the use of these plastic insulated screws, we would appreciate your supplying us with their full address.—J. C. HART
Canadian Radio Manufacturing Corp., Ltd.
Leaside, Ontario

Ed—The full address is Forman Insulating Screw Corporation, 401 Broadway, New York 13, N. Y.

RAYTHEON

fractional hp motors

dependable performance



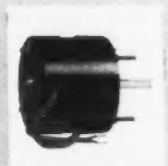
TYPE #470 6-pole, 1050 rpm, 1/15 to 1/8 hp. New economy model featuring drawn steel end bells with self-aligning bearings. Helps you cut production costs—for example you can get the Type #470 at about ½ the cost of comparable capacitor motors, or ⅔ the cost of split phase motors. Typical example of Raytheon "New Principle" design and Creative Electronic Craftsmanship. Write for Bulletin #4000.

"Count on Raytheon to run it." Design engineers have learned to rely on Raytheon fractional hp motors when they want to *run* things—for example, FANS, BLOWERS and AIR CONDITIONING EQUIPMENT.

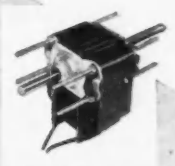
Raytheon Creative Craftsmanship assures *Dependable Performance* at the shaft—the kind of trouble-free power unit that manufacturers need in today's competitive markets.

Call in your Raytheon representative for consultation on your specific application.

RAYTHEON Type 230 Shaded 2-pole 1/200 to 1/50 hp at 3200 rpm motor.



RAYTHEON Type 330-S Shaded 4-pole 1/200 to 1/10 hp 1550 rpm motor.



RAYTHEON Type 350 Shaded 2-pole, 1/40 to 1/6 hp at 3000 rpm motor.

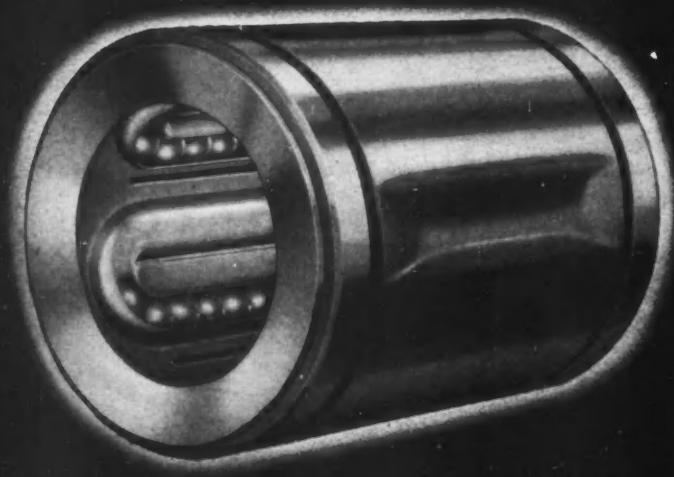
RUSSELL ELECTRIC COMPANY

MOTOR DIVISION OF RAYTHEON MANUFACTURING CO.

4501 So. Western Boulevard Dept. K-22 Chicago 9, Illinois

Announcing A New LOW COST Ball Bearing for Your

LINEAR MOTIONS



BALL BUSHING

THE COMMERCIAL GRADE SERIES B

Sliding linear motions are nearly always troublesome. Thousands of progressive engineers have solved this problem by application of the Precision Series A Ball Bushing.

The low-cost Commercial Grade Series B bearing is now added to the Ball Bushing line and offered to original equipment manufacturers. This ball bearing has been developed for support of linear motions in competitively priced, volume produced products where super precision is not essential. Alert designers can now make tremendous improvements in their products by using Ball Bushings on guide rods, reciprocating shafts, push-pull actions, or for support of any mechanism that is moved or shifted in a straight line.

Competition is returning. Up-to-date engineering can be *important to you!*

- LOW FRICTION
- ELIMINATE BINDING AND CHATTER
- SOLVES SLIDING LUBRICATION PROBLEMS
- LASTING ALIGNMENT
- LOW MAINTENANCE
- LONG LIFE

NEW 1½" NOW AVAILABLE

Now in production for ¼", ½", ¾", 1" and 1½" shaft diameters. If you have a catalog, phone your representative or write us for new Data Sheet. If not, write for complete literature and the name of our representative in your city.

THOMSON INDUSTRIES, INC.

Dept. F 〉 *MANHASSET, NEW YORK*

PROGRESSIVE MANUFACTURERS USE BALL BUSHINGS — A MAJOR IMPROVEMENT AT A MINOR COST

NEW BOOKS

Elementary Pile Theory

HARRY SOODAK *and* EDWARD C. CAMPBELL, 73 *pages,* 8½ x 5½ *in. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York* 16, N. Y. $2.50

It is expected that few readers of PRODUCT ENGINEERING will need to become experts on the construction and use of chain-reacting piles as part of their regular jobs. For those, however, who like to keep up with the latest developments in the field of physics and applied science, this book will give a good basic understanding of an important new field.

The book uses simple mathematics to explain the basic fundamentals. There is more advanced mathematics for those who are interested in a further explanation of any one of the main parts.

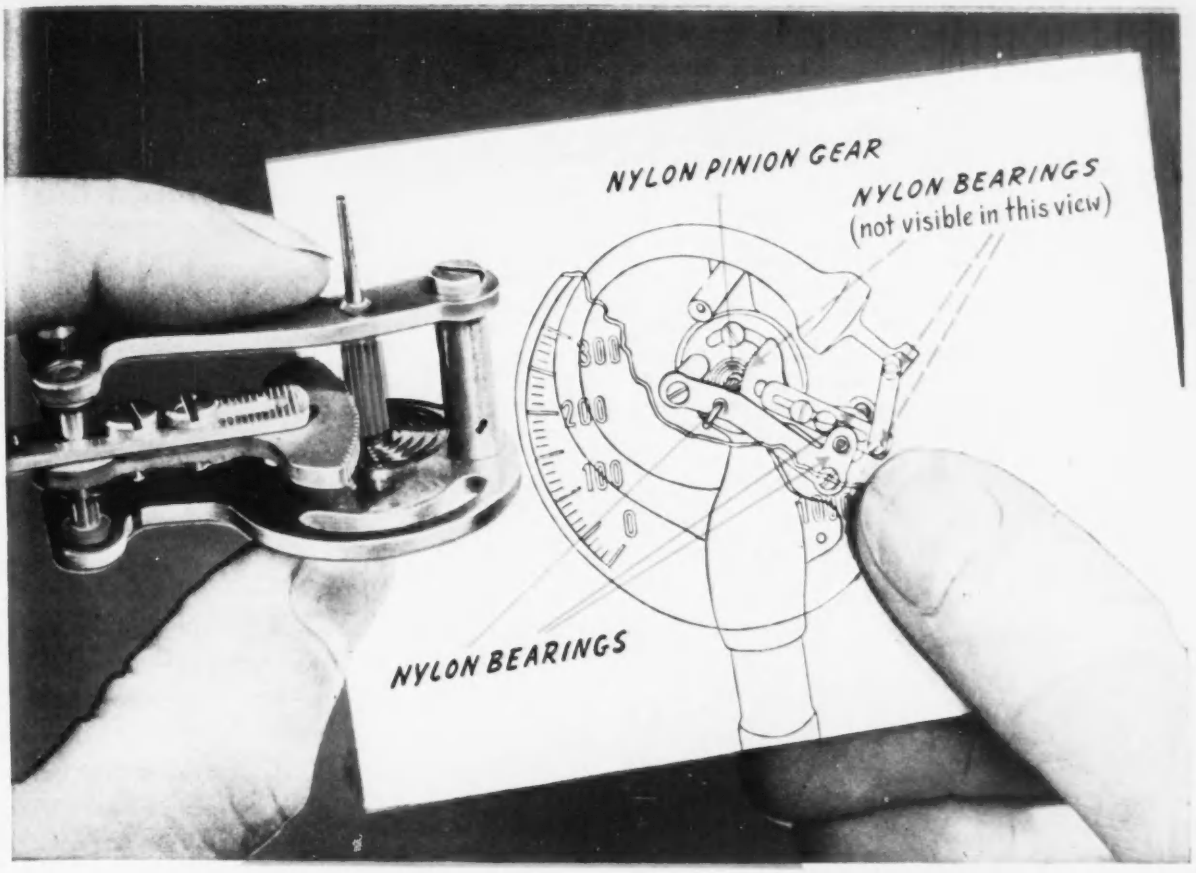
The authors report on detailed investigations of what happens to the fast neutrons given off in the act of fission, how they move about in space and how they are slowed down in matter. To avoid complications, only homogeneous piles are treated, and they are considered from the point of view of the simple diffusion theory.

Principles of Stretch-Wrap Forming

87 *pages* 8½ x 11 *in. Published by Hufford Machine Works, Inc., 207 North Broadway, Redondo Beach, Calif.* $5

The stretch-wrap forming process has been developed since the entry of the United States into World War II, and until now there has been no technical literature on the subject. This book was prompted by Hufford's findings that among users of their equipment there were a wide diversity of tooling techniques and many misconceptions concerning the stretch-wrap forming process. This book attempts to clarify many ideas and to include as much tooling information and as many actual operating examples as are available at present. It presents the theory, advantages and uses of stretch-wrap forming equipment in modern aircraft manufacture, together with detailed tooling procedures and form die construction data.

Briefly, the process is as follows: A die possessing the desired contour is securely mounted and remains im-



NYLON PLASTIC GEARS AND BEARINGS

ABSORB SHOCK...RESIST WEAR...GIVE LONG LIFE

Severe vibrating and pulsating will wear down the mechanism of an ordinary pressure gauge . . . cause it to register inaccurately. To produce a gauge that would operate with utmost efficiency over extended periods,

NYLON PINION is machined from blank injection-molded directly onto metal shaft by Plastic Mfrs. Corp., Stamford, Conn. Nylon bearings are machined from rod stock supplied by the Polymer Corp., Reading, Pa. Accuracy of the gauge is guaranteed by its manufacturer to within one half of one per cent of dial graduation over entire scale of a full 270° dial arc.

Manning, Maxwell and Moore developed a "nylon movement" for their Ashcroft "Duragauge." Five years of extensive field testing prove that the nylon movement outperforms and outlasts other gauge movements under severe conditions of vibration, pulsation and fretting corrosion.

Nylon parts offer an extremely low coefficient of friction. Too, nylon is highly resilient . . . resists abrasion. It absorbs vibrations without passing them along. Under shock loading, nylon's "plastic memory" allows parts to return to original size . . . maintains accuracy.

A large chemical company installed one of these gauges on "the worst pulsating service to be found." After 10 months of continuous use, the gauge was removed. Under microscopic examination, the nylon movement showed no visible wear!

Durable nylon plastic is finding more and more uses in gears, bearings, and machine parts for textile machinery, business machines, movie cameras, electric shavers, household appliances, and many other types of equipment. Its properties may well help you, too. For literature on nylon and other Du Pont plastics, write today:

E. I. du Pont de Nemours & Co. (Inc.)
Polychemicals Dept., Sales Offices:
350 Fifth Avenue, New York 1, N. Y.;
7 S. Dearborn St., Chicago 3, Ill.; 845
E. 60th Street, Los Angeles 1, Calif.

DU PONT
REG. U. S. PAT. OFF.
PLASTICS
Better Things for Better Living
. . . through Chemistry

movable during the forming process. The work is held in tangency to the front of the die and stretched almost to its yield point. With tension maintained, the work is wrapped around the die, accurately molding itself into and around each die configuration. Upon completion of the wrap, another stretch is applied which slightly exceeds the elastic limit of the workpiece. This second stretch "sets" the contour shape and simultaneously raises the yield strength of the material. Upon release, the part retains the shape imparted by the die curves. The process is simple, fast and effective.

Typical chapters include: Theory of Stretch-Wrap Forming; Actual Production Case Histories; Hufford Machine Specifications; Selecting the Proper Stretch-Wrap Forming Machine; Examples of Stretch-Wrap Forming; Laboratory Reports.

It is believed this treatise will be of great help to all foremen and superintendents in metalworking departments of aircraft plants, as well as to their executives. Courtesy copies will be sent to all qualifying personnel. The book, however, is not for general distribution.

Physics, Volume I and II

GEORGE SHORTLEY, *Professor of Physics, Ohio State University and* DUDLEY WILLIAMS, *Assoc. Professor of Physics, Ohio State University.* 1,271 *pages, total,* 6½ x 9¼ *in. Published by Prentice-Hall Inc., 70 Fifth Ave., New York, N. Y. Volume I.* $6 *and Volume II.* $7.35

This is the most complete volume of physics that has come to our attention in the last few years. The authors state that it is intended for the use of sophomore college students. Without much skipping and deletion, however, we are sure that these two volumes are far too extensive in their treatment for an ordinary sophomore physics course. The authors have indicated which chapters can be skipped, or taught in part only, to bring the book down to the scope of the usual college course. The rest of the material can be used by advanced college students or as a reference for graduate engineers and physicists.

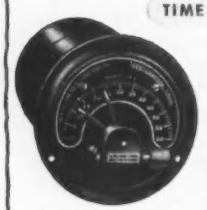
The engineer who took a standard course in physics several years ago will be amazed to learn of the strides made in that field and will be surprised to see them so well presented in one complete work. Students will be glad to see that this book has been written so that nothing will have to be "unlearned" when he takes the advanced

PRECISION
TIME CONTROL
by
Cramer

PERCENTAGE TIMERS

Automatically control the percentage of time at which circuit can be closed or opened out of a definite length of time. Adjusted from the front, can be set from 5% to 197% of total cycle. Write for Bulletin 2500G.

TIME DELAY RELAYS

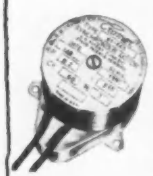


Provides adjustable or fixed time delay between closing and opening of a load circuit. Has automatic re-set feature which restores timer to its normal starting position when control circuit fails or is de-energized. Also available with reverse clutch which prevents re-set in case of power failure. Write for Bulletin 800H.

RUNNING TIME METERS

Synchronous motor driven, automatically and cumulatively register the total operating time or idle time on any circuit, machine or system. Also available with re-set feature. Write for Bulletin 3500E.

SYNCHRONOUS MOTORS



Efficient, powerful, self-starting synchronous motors for applications requiring constant speed at given frequency. Compact...precision-built... available with output speeds from 60 RPM to 1/24 RPH. Write for Bulletin 10D.

If the performance of your product depends on precision timing, send us full particulars about your application. In developing a range of devices for a great variety of commercial and industrial timing duties, we have had wide experience in the design of equipment to meet many specific needs... and will welcome the opportunity to assist in solving your problem.

THE R. W. CRAMER COMPANY, INC.
Box 7, Centerbrook, Connecticut

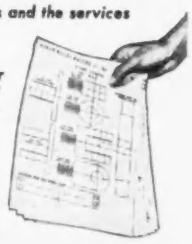
SPEED PRODUCTION WITH
10-DAY
DELIVERY
SPEED DESIGN WORK WITH
FREE
TEMPLATES
SAVE SPACE WITH
O-M
CYLINDERS

Ortman-Miller cylinders for air, hydraulic and water applications are available in the full range from 1½" to 8" bores on *ten-day delivery!* Add this to the other advantages of space saving and simplicity of design (no tie-rods, no bulky end caps).

All body parts machined steel (not cast), all bearing surfaces bronze. Circumferential keys allow disassembly and repacking in a matter of minutes, or interchangeable mounting brackets may be fitted without disassembly. Parts adjustable to any angle.

Write today for details and the services of your distributor.

And... write today for free complete set of templates in one-half and full scales. Precision drawn to save your time, speed design work. Shows all O-M cylinders and all mounting devices.



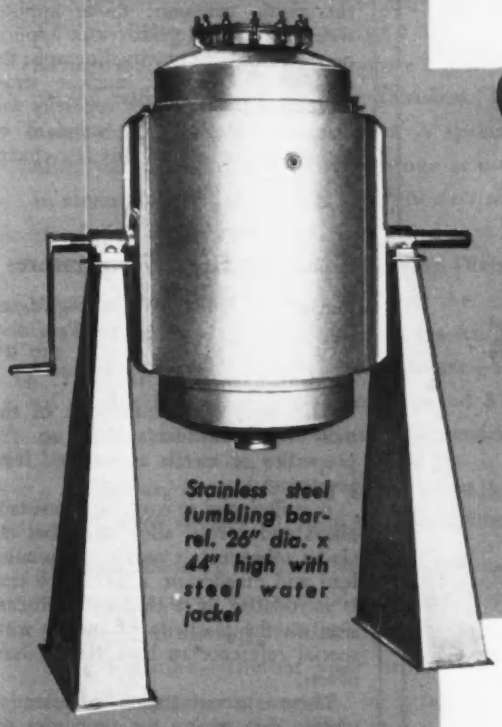
ORTMAN-MILLER
MACHINE CO., INC.
1218 150th Street, Hammond, Indiana

BACK IN THE 1880's

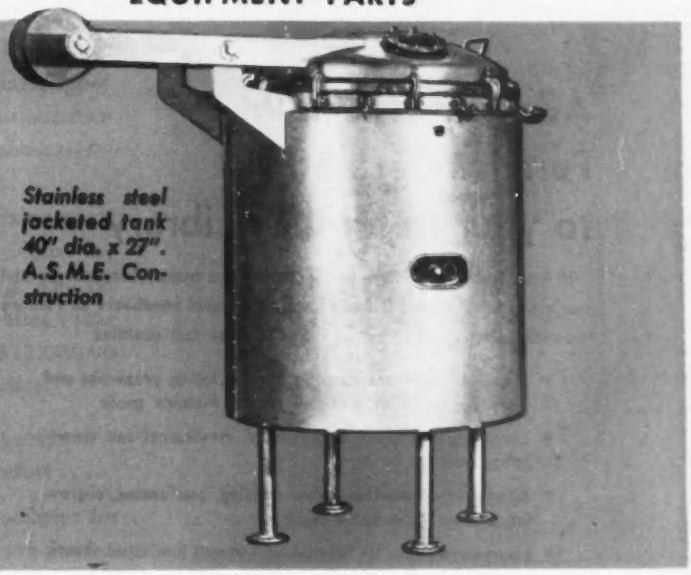
WHEN WATERMAN DEVELOPED THE FOUNTAIN PEN

Koven

WAS MAKING INDIVIDUALIZED EQUIPMENT PARTS



Stainless steel tumbling barrel. 26" dia. x 44" high with steel water jacket

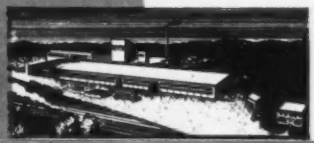


Stainless steel jacketed tank 40" dia. x 27". A.S.M.E. Construction

L. O. KOVEN & BRO., INC.
154 Ogden Ave., Jersey City 7, N.J.

Plants:

Jersey City, N. J. Dover N. J.



The history of KOVEN's engineering skill goes back to the epic eighties, when our country made great strides in the inventive field. Always a leader, KOVEN has specialized for over 69 years in the fabrication of individualized parts for many of the nation's leading equipment manufacturers. Our vast facilities and perfected methods make it possible for us to turn out the specific part or parts you require in less time and often at a lower cost than could be done in your own plant. Call or write for a consultation with a KOVEN representative now — no obligation.

In our two huge plants are complete, modern facilities which include machine, welding, plate and sheet metal, galvanizing and painting shops. X-ray equipment insures quality control.

KOVEN FOR INDIVIDUALIZED EQUIPMENT PARTS SINCE 1881

PRODUCT DESIGNERS!

Add this Valuable Formica Data Book to your Reference Library

It won't stay there long—in the library, we mean—because you and your associates will find it one of the most useful handbooks on Formica laminated plastics. This authoritative 96-page text contains

- A separate and complete data sheet listing properties and performance characteristics for every Formica grade
- Compares each grade for electrical, mechanical and chemical properties
- Helpful information on special molding, postforming, engraving and rubber surfaced stock
- Recommendations for fabricating Formica laminated sheets, tubes and rods
- Military specifications for armed services

Printed in reference book style and carefully indexed. It's durably bound in loose-leaf binder so that you can keep your copy up-to-date. Priced at $1.00 a copy—worth many times the money. Send a dollar today for your personally numbered copy.

THE FORMICA CO.
4676 Spring Grove Ave.
Cincinnati 32, Ohio

Productive FORMICA at Work in Industry

physics courses. The authors use one set of terminology in speaking of the gravitational system and one absolute system of mechanical units when speaking of electromagnetic subjects. Since both of the standard systems used are encountered in more advanced work, there will be no confusing transfer to unfamiliar systems of measurement.

Since the book is written for the use of students who will be beginning their mathematics course concurrently with the physics course, the author starts with plane trigonometry and gradually increases the complexity of mathematical explanations.

All of the subjects usually encountered in a physics course are covered in great detail, with a complete chapter given over to each of the usual physical subjects. In addition to the 42 chapters covering the standard subjects, the last three chapters deal with electronics, electromagnetic waves, and elementary particles. Such subjects as thermionic emission; diode applications; the control grid; triode applications; the electron-ray oscillograph; the electron microscope; nuclear structures; and the theory of relativity will bring the present day standard of physics up to date on this fast changing subject.

Metals at High Temperatures

FRANCES H. CLARK, *Consulting Metallurgist, 372 pages, 6 x 9 in. Published by Reinhold Publishing Co., 330 W. 42nd St., New York 18, N. Y.* $7

This book is a compilation of the most recent available data on the properties of metals at elevated temperatures.

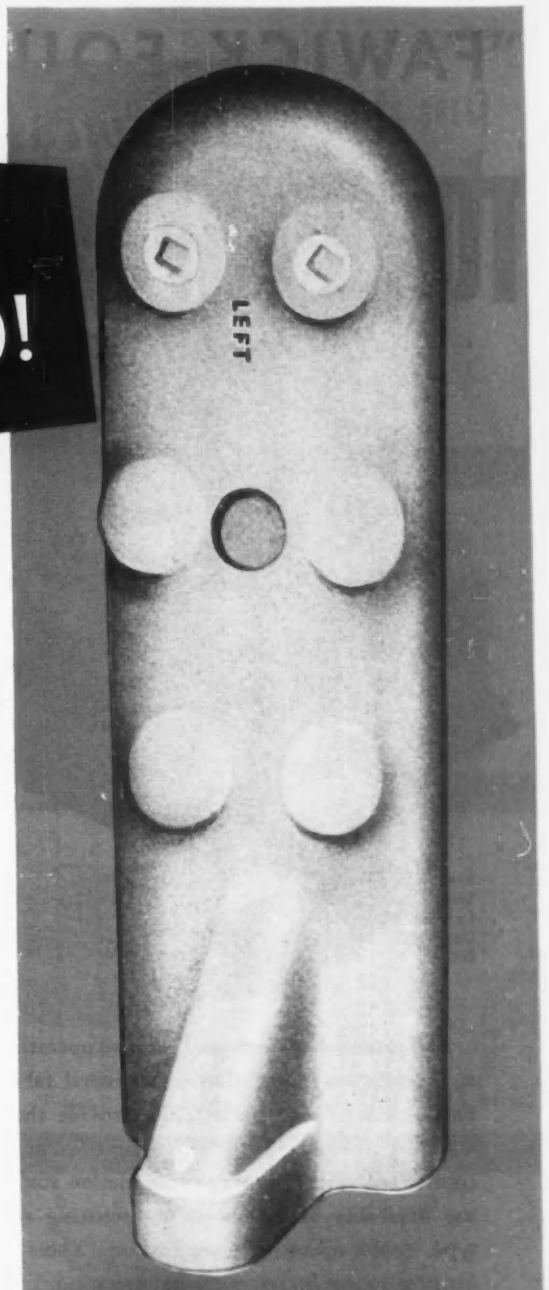
It is designed to cover heat resistant alloys and special alloy steels but includes other metals such as aluminum, lead and magnesium alloys. The text is introduced by a theoretical discussion on the plasticity of metals with special reference to high temperature effects.

There is a vast amount of essential tabular data giving the properties of all the latest alloys which have been developed for service at extremely high temperatures.

A full discussion is devoted to the phenomenon of creep in metals and alloys, and an effort has been made to include all the most relevant data on this important property. There is also a section on test methods for metals at high temperature and current manufacturing methods for heat resistant alloys.

This book is intended for all who are concerned with the engineering

THIS CASTING SAVED $12,000.00!



Why an Electrode Holder, redesigned by National Bearing Division, lasted over 73 times longer

A large Midwest plant was getting 5 days service from furnace electrode holders . . . before a trial order was placed with National Bearing Division. This plant saved dollars—*right from the start.* A special copper alloy permitted sound, dense castings at no sacrifice in electrical conductivity. Result? *30 days service—instead of 5—before replacement.*

On the next order, National Bearing Division engineers submitted a new design for the clamp—for greater strength and resistance to cracking. The redesigned electrode holders are still in use—after a year of trouble-free operation! They've saved $12,000.00 on replacement costs and furnace maintenance.

Better, longer-lasting non-ferrous parts can save money in your plant or product . . .

National Bearing Division has complete facilities for finding practical, economical solutions to non-ferrous bearing and casting problems. Investigate these complete facilities—whether your problem requires designed engineering service, or production of non-ferrous parts to your own specifications.

If you design, specify or purchase . . . This catalog will interest you

Find out how complete facilities for Research, Engineering and Production can reduce costs of non-ferrous parts in your plant or product. Write for a free copy of this catalog.

NBM

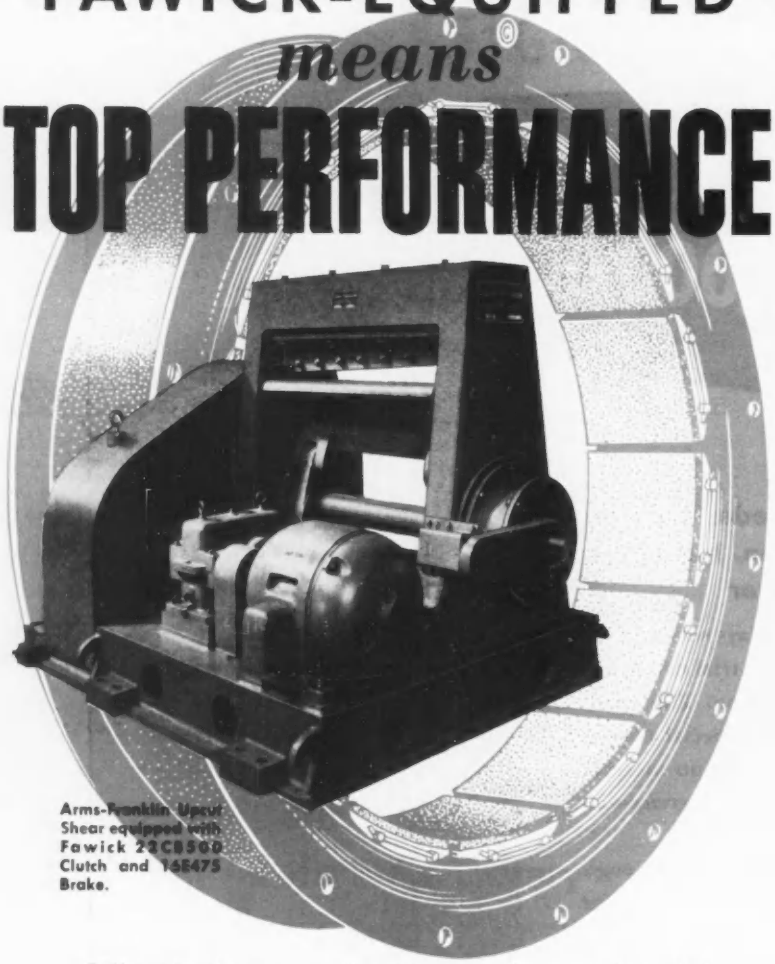
AMERICAN Brake Shoe COMPANY

NATIONAL BEARING DIVISION

4932 Manchester Avenue • St. Louis 10, Mo.

PLANTS IN: ST. LOUIS, MO. • MEADVILLE, PA. • NILES, OHIO • PORTSMOUTH, VA. • ST. PAUL, MINN. • CHICAGO, ILL.

"FAWICK-EQUIPPED"
means
TOP PERFORMANCE



Arms-Franklin Uncut Shear equipped with Fawick 23CB500 Clutch and 16E475 Brake.

Full productive power and positive operating control are vital to top-efficiency operation of all metal fabricating machines. Fawick Clutch and Brake units provide these machines with every control factor necessary to top-efficiency production: 360° constant-velocity contact at the friction surface; shock-absorbing flexibility of the torque-transmitting rubber tube; open-type, cooler-operating construction. These exclusive Fawick features insure better machine operation.

Add to these the self-adjustment and low maintenance characteristics of Fawick units and you have the answer to your clutch and brake problems—FAWICK.

For detailed information on all advantages of Fawick Clutch and Brake units, write to the Main Office, Cleveland, Ohio, for Bulletin 300.

All desirable clutch characteristics are built into Fawick Airflex units

FAWICK Airflex CO., INC.
9919 CLINTON ROAD CLEVELAND 11, OHIO

FAWICK CLUTCHES = = PEAK EFFICIENCY
DUAL CONTACT 360° RADIAL CONTACT SHOCK ABSORPTION CONTROLLED TORQUE

and metallurgical aspects of high temperature service, particularly in reference to turbojet airplane engines where these alloys are playing a critical part. Numerous other high temperature demands are being made upon alloys in military uses, such as those for rockets, projectors and gun barrels. All metallurgists and mechanical engineers will welcome its abundance of easy-reference information.

Resistance Welding, Designing, Tooling And Applications

WALLACE A. STANLEY, *Consultant on Product Design, Tooling and Manufacturing Operations, Progressive Welder Co., 320 pages, 8½ x 11 in. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York 18, N. Y. $7.50.*

An engineering handbook meets reference needs from as many angles as possible with the maximum amount of specific information for each man. The above book could almost be classified as a resistance welding handbook. The field is comparatively new and the book seems more qualitative than numerical, but at least one of the sections included should be of interest to every design, cost or production engineer in a plant using or contemplating resistance welding.

The history and general principles are covered first, in a pre-digested lesson form that is especially valuable to the novice. Resistance welding classifications are defined, and the reader gets a good basic knowledge of the electrical circuits involved. Line illustrations scattered through the text reinforce each point, while large photographs are used merely for background. Stress analysis is glanced at to explain the general approach required by resistance welding, while the mathematics involved are left to text books. Costs are explained, and resistance welding is balanced against other joining methods.

The sections on resistance welding machines will interest production men. Standard, portable and special purpose machines differ in production costs, capabilities and applications. The special production rigs shown are perhaps of more value as thought-provokers than for imitation.

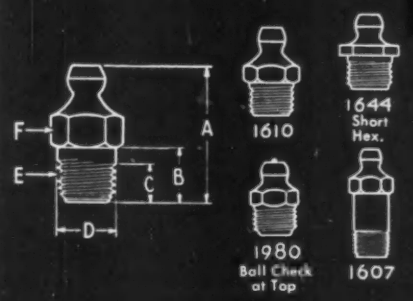
Two approaches to welding design are discussed. Resistance welding can replace other fasteners where economy dictates on a finished design, or the part itself can be redesigned for resistance welding. The author votes for the purist method of complete

Genuine ALEMITE LUBRICATION FITTINGS

DATA SHEET **NO. 1**

Simplify ORDERING, INSPECTION and INVENTORY CONTROL. Specify Alemite part numbers on your blueprints.

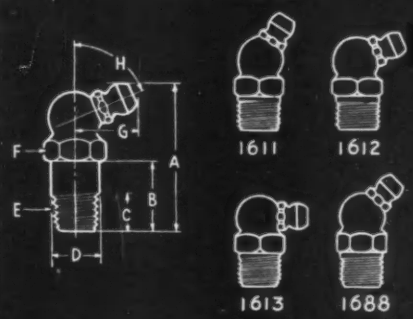
Useful Information to Help You Select and Specify the Right Alemite Fitting for Every Bearing



STRAIGHT HYDRAULIC FITTINGS

Part No.	A	B	C	D	E†	F
1607	1 17/64"	25/32"	13/64"	13/32"	1/8" P.T.	7/16"
1610	3/4"	17/64"	13/64"	25/64"	1/8" P.T.	7/16"
1618	1 7/64"	5/16"	1/4"	1/2"	1/8" P.T. (Female)	1/2"
1634	13/16"	5/16"	1/4"	13/32"	1/8" P.T.	7/16"
1644	21/32"	5/16"	1/4"	13/32"	1/8" P.T.	7/16"
*1650	57/64"	3/8"	19/64"	13/32"	1/8" N.P.T.F.	7/16"
1669	1 3/4"	1 17/64"	1/4"	13/32"	1/8" P.T.	7/16"
1684	2 5/8"	2 3/16"	5/16"	13/32"	1/8" P.T.	3/8"
*1708	53/64"	23/64"	19/64"	13/32"	1/8" N.P.T.F.	11/32"
1980	41/64"	17/64"	7/32"	13/32"	1/8" P.T.	7/16"

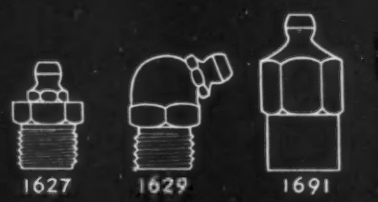
*Leak-proof seal
†Thread size listed as 1/8" P.T. is 1/8" P.T.F.—SAE EXTRA SHORT



1/8" THREAD ANGLE HYDRAULIC FITTINGS

Part No.	A	B	C	D	E†	F	G	H
1606	1 13/16"	1"	3/8"	13/32"	1/8" P.T.	7/16"	15/32"	90°
1611	1 1/64"	17/64"	13/64"	25/64"	1/8" P.T.	7/16"	21/64"	30°
1612	53/64"	17/64"	13/64"	25/64"	1/8" P.T.	7/16"	1/2"	67 1/2°
1613	49/64"	17/64"	13/64"	25/64"	1/8" P.T.	7/16"	33/64"	85°
1614	1 5/64"	5/16"	1/4"	13/32"	1/8" P.T.	7/16"	31/64"	105°
1620	1 5/16"	13/32"	5/16"	7/16"	1/8" P.T. (Female)	7/16"	17/32"	45°
1623	1 3/16"	9/16"	1/4"	13/32"	1/8" P.T.	7/16"	31/64"	67 1/2°
1638	2 9/32"	1"	3/8"	13/32"	1/8" P.T.	7/16"	21/64"	30°
1649	2 25/32"	2 3/16"	5/16"	13/32"	1/8" P.T.	3/8"	29/64"	67 1/2°
*1651	1"	3/8"	19/64"	13/32"	1/8" N.P.T.F.	7/16"	1/2"	67 1/2°
1688	61/64"	17/64"	13/64"	25/64"	1/8" P.T.	7/16"	27/64"	45°
*1692	1 7/32"	3/8"	19/64"	13/32"	1/8" N.P.T.F.	7/16"	21/64"	30°
*1693	15/16"	3/8"	19/64"	13/32"	1/8" N.P.T.F.	7/16"	31/64"	90°
*1950	1 1/8"	3/8"	19/64"	13/32"	1/8" N.P.T.F.	7/16"	15/16"	105°

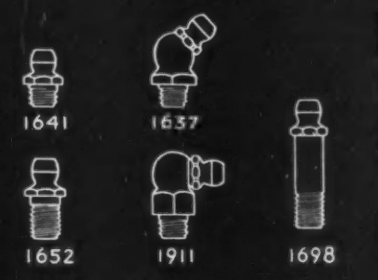
*Leak-proof seal. †Thread size listed as 1/8" P.T. is 1/8" P.T.F.—SAE EXTRA SHORT



1/4" PIPE THREAD HYDRAULIC FITTINGS

Part No.	A	B	C	D	E†	F	G	H
1627	19/32"	23/64"	9/32"	17/32"	1/4" P.T.	9/16"	.....	.....
1629	1"	11/32"	9/32"	17/32"	1/4" P.T.	9/16"	3/8"	67 1/2°
1691	1 21/64"	Female Thread			1/4" P.T.	3/8"	.....	.....

†Thread size listed as 1/4" P.T. is 1/4" P.T.F.—SAE EXTRA SHORT.



1/4"—28 THREAD HYDRAULIC FITTINGS

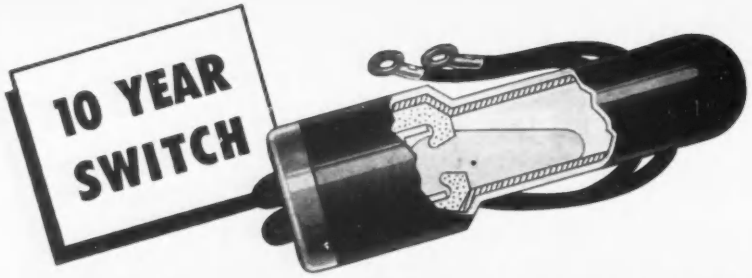
Part No.	A	B	C	D	E	F	G	H
1636	31/32"	19/64"	1/4"	17/64"	1/4"-28 Taper USF-Thd.	3/8"	3/8"	45°
1637	27/32"	11/64"	1/4"	1/4"	1/4"-28 Taper USF-Thd.	3/8"	3/8"	45°
1641	31/64"	11/64"	1/4"	1/4"	1/4"-28 Taper USF-Thd.	5/16"	.....	.....
1652	43/64"	23/64"	1/4"	1/4"	1/4"-28 Taper USF-Thd.	5/16"	.....	.....
1680	59/64"	39/64"	1/4"	17/64"	1/4"-28 Taper USF-Thd.	5/16"	.....	.....
1698	1 7/64"	51/64"	1/4"	17/64"	1/4"-28 Taper USF-Thd.	17/64"	.....	.....
1703	1/2"	7/32"	5/16"	1/4"	1/4"-28 Taper USF-Thd.	17/64"	.....	.....
1770-A	19/32"	1/8"	7/64"	1/4"	1/4"-28 NF-2-Thd.	3/8"	25/64"	45°
1792	31/64"	9/64"	1/8"	1/4"	1/4"-28 NF-2-Thd.	17/64"	.....	.....
1911	47/64"	11/64"	1/4"	1/4"	1/4"-28 Taper USF-Thd.	3/8"	15/32"	90°

ALEMITE

Modern Lubrication Methods That Cut Production Costs

FREE! Complete set of four Data Sheets similar to this, giving specifications on all types and sizes of Alemite Lubrication Fittings. Simply write to Alemite, Dept. T-110, 1850 Diversey Parkway, Chicago 14, Illinois.

Another Product of STEWART WARNER



TROUBLE-FREE

for the Life of Your Product

- ★ Seven new models—1 to 65 amperes
- ★ Millions of contacts assured
- ★ Smaller in size—increased in capacity
- ★ Positive "Make and Break" contact action
- ★ No deterioration from inactivity
- ★ Moisture proof if desired
- ★ Withstands high temperatures

Send for BULLETIN 525

Steel **Durakool** MERCURY SWITCHES

See Telephone Directory for Local distributor or write
DURAKOOL, INC. • ELKHART, INDIANA

SMALL
PARTS
Cost less when made by
MULTI-SWAGE
The economy way to get a million small parts similar to these —

Examine the tubular metal parts shown here twice size. If you use anything similar . . . in quantities of over a million . . . important savings can be yours. Send us the part and specs. Our quotation will show why the Bead Chain Company's MULTI-SWAGE Process has long been known as the most economical method of making electronic contact pins, terminals, jacks and sleeves. And, why more and more users of mechanical parts (up to ¼" dia. and to 2" length) employ our facilities. WRITE for Data Bulletin.

THE BEAD CHAIN MANUFACTURING CO.,
Tr. Mark 92 MOUNTAIN GROVE ST., BRIDGEPORT 5, CONN.

design reappraisal. Spot, seam, projection and flash welding design are considered in detail, with the individual advantages of each. The limitations placed on design by shop tools and the methods are clearly illustrated.

In summing up, the author covers the production problems of testing, scheduling, materials handling and tooling. This section is especially important to plants considering new or expanded resistance welding operations and applications.

Fundamentals of Calculus

DONALD E. RICHMOND, *Professor of Mathematics at Williams College, 233 pages, 6 x 9 in. Published by the McGraw-Hill Book Co., 330 W. 42nd St., New York 18, N. Y., $3.*

This textbook is intended to provide sufficient knowledge of calculus to furnish an adequate background for courses in physics using exponentals and trigonometric functions. It is not strictly an engineering text but could be used, for example, by liberal arts students who elects one year of college mathematics to acquire some feeling for mathematical thinking. The material is presented with as much attention to logical clarity as possible.

Particular emphasis is placed throughout on fundamental principles with problems specially designed to contribute to the student's understanding. The text assumes no knowledge of analytic geometry, advanced algebra or analytic trigonometry, but develops what is essential in these subjects in close connections with the rest of the text.

The exercises in the book were designed expressly to promote an understanding of the text. The text was not written to teach the student how to do the problems. The problems are a means to an end—the understanding of a way of thinking—but they are an essential means to this end. In many cases, the student is asked in the exercises to extend and supplement the theory for himself.

Steam Turbines and Their Cycles

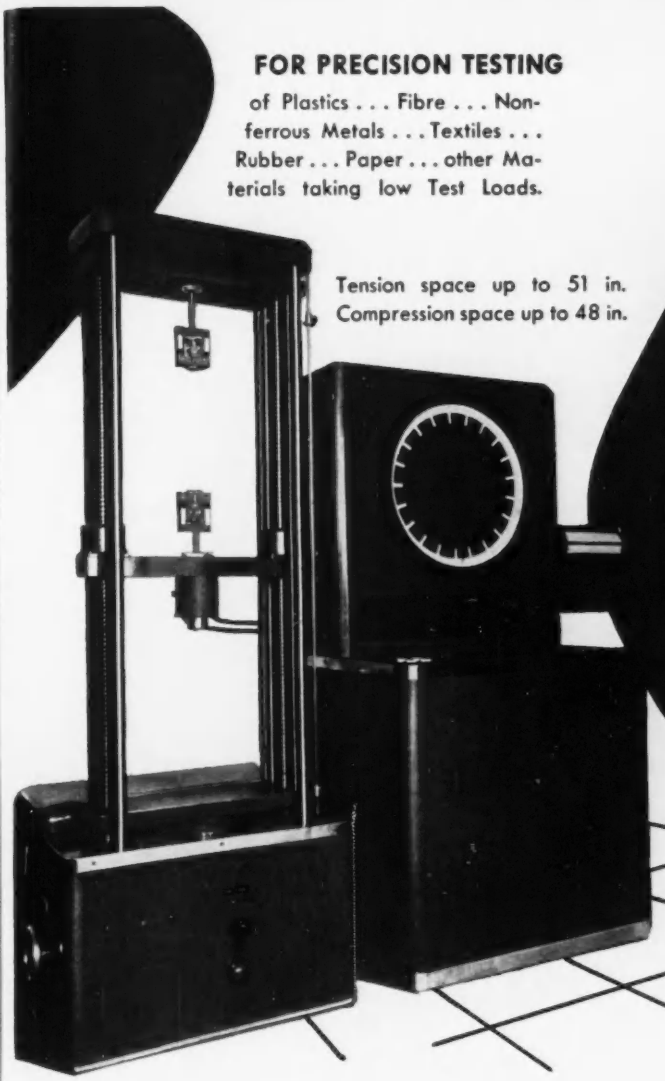
J. KENNETH SALISBURY, *Division Engineer of the Thermal Power Systems Division, General Electric Co., 645 pages, 6 x 9 in. Published by John Wiley & Sons, Inc., 440 Fourth Av., New York 16, N. Y. $9.*

The essentials of steam turbine design are presented for the guidance of users and manufacturers of steam tur-

FOR PRECISION TESTING

of Plastics . . . Fibre . . . Non-ferrous Metals . . . Textiles . . . Rubber . . . Paper . . . other Materials taking low Test Loads.

Tension space up to 51 in.
Compression space up to 48 in.



NEW DESIGN—
NEW FEATURES

BALDWIN

TATE EMERY P-T-E UNIVERSAL TESTING MACHINE

This important machine projects the sensitivity, accuracy and usefulness of the Baldwin standard line into the low-test-load field. Available in standard 5000-lb. capacity, with ranges of 5000, 1000, 200 and 50 lbs.; furnished in lower capacities when necessary. Many new features: solenoid-operated range changer, flexibility of cabinet location, convenient access doors, rugged construction, high intensity indirect dial lighting. Automatic cycling available as an extra.

EMERY HYDRAULIC CAPSULE. The weighing system utilizes the famous Emery Hydraulic Capsule—no fluid flow, no volume change, no friction, negligible inertia; load-pressure curve is always linear.

TATE-EMERY AIR CELL. Applying an Air Cell provides two extra low ranges—10 lbs., and 2 lbs. for tension and compression testing. Load measured on unique pneumatic system. Accuracy ¼% of reading or 0.15% of either range capacity.

CONVENIENT CONTROLS. Selector switch for range change, 400-1 range testing speed rheostat, and other controls all handily grouped on control panel.

RECORDERS. Baldwin Stress-Strain Recorders readily attached; operate on any load range. (Ask for Bulletin 262 for complete recorder information.)

Bulletin 305 gives full information on the P-T-E machine. Ask the nearest office for a copy—or write us direct.

The Baldwin Locomotive Works, Philadelphia 42, Pa., U. S. A. Offices: Chicago, Cleveland, Houston, New York, Philadelphia, Pittsburgh, San Francisco, St. Louis, Washington. In Canada: Peacock Bros., Ltd., Montreal, Quebec.

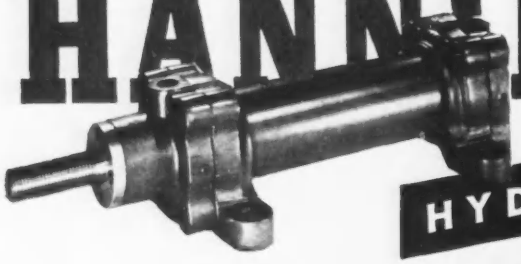
BLW THE BALDWIN GROUP

BALDWIN

TESTING HEADQUARTERS

Here's Why it **PAYS** *to Use*

HANNIFIN



HYDRAULIC

PNEUMATIC

CYLINDERS

1. **QUICKER DELIVERY!** You can get cylinders to meet virtually any need out of the Hannifin standard line. Save time and money right from the start by checking with Hannifin on all of your cylinder requirements.

2. **EVERYTHING YOU WANT!** Wide choice of mounting styles. Any length stroke. Adjustable cushions for head end, rod end, or both. Single or double end rods. Air Cylinders in standard sizes from 1" to 16"; Hydraulic Cylinders, 1" to 8". Larger or special types to order. "Hy-Power" cylinders for pressures to 5,000 p.s.i.

3. **SUPERIOR DESIGN AND CONSTRUCTION!** A precision-built product; latest, most advanced design features; unexcelled quality.

4. **WORK BETTER! LAST LONGER!** The choice of engineers and machinery builders for more than 40 years. Less trouble, less maintenance. For recommendations, see your local Hannifin representative, or write.

HYDRAULIC CYLINDERS
Bulletin 110

ASK FOR BULLETINS

Hannifin CORPORATION
1125 S. Kilbourn Ave.
Chicago 24, Illinois

AIR CYLINDERS
Bulletin 210

bines, designers of power plant cycles and turbines, and test engineers in power plants.

Of particular value is the author's discussion of a new method for cycle evalution. This method permits the simple and accurate calculation of losses in the feedwater-heating cycle. The method also facilitates direct correction from test to guarantee cycle, economic appraisal of proposed design changes in a given cycle, and a perspective of losses in feedwater heating systems.

Portions of the book are an extension of a paper, "The Steam Turbine Regenerative Cycle—An Analytical Approach," presented before the ASME, for which the author was awarded the Melville Medal.

The twenty-six chapters of the book are grouped under four major classifications: Fundamentals of turbine design; the regenerative cycle; cycle analysis; and application engineering. Data on steam, theoretical heat rates, and steam charts are included in the Appendix for ready reference. Numerous illustrations are used, and practical problems are included at the end of each chapter.

Twelve pages of the book are devoted to a discussion of gas turbine cycles, a subject that is now of considerable economic importance and which no longer can be classified as of only academic interest.

High Speed Computing Devices

C. B. TOMPKINS *and* J. H. WAKELIN. *Edited by W. W. Stifler, Jr.*, 451 *pages*, 6 x 9 *in. Published by the McGraw-Hill Book Co.*, 330 *W. 42nd St., New York* 18, *N. Y.* $6.50

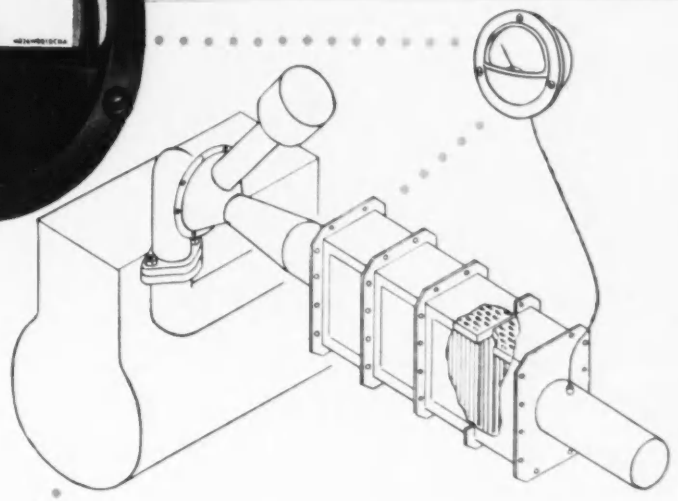
This timely book acquaints the reader with the various mathematical methods and the physical mechanisms which have been developed for use in automatic computation.

Straightforward in its approach, the material presented has been confined to methods which already exist or which seem likely to be available shortly.

You are introduced to the general character of computing machines; the arithmetic techniques employed in using them are clearly outlined, and basic circuits used to perform various computing-machine functions are described. The book points out varied uses to which large-scale machines can be put and explains the importance and nature of the relationship between a machine and its human operator who supplies mathematical ingenuity and instructions in language the computing

MARION . . . helps the OXY-CATALYST Mfg. Co.

MAKE EXHAUST GASES NON-POISONOUS



OCM manufactures a new type of Catalytic Exhaust which takes the place of the standard muffler . . . makes the exhaust gases of gasoline engines both non-poisonous and odorless. When applied to Fork Lift Trucks, for example, it makes them safe for inside use.

In developing the OCM Exhaust, a special type of ruggedized pyrometer was needed to withstand the deteriorating abuse to which OCM equipped machines are subjected. Critical accuracy, and exceptional dependability under extreme conditions of shock and vibration were necessary. Only Marion's new line of *ruggedized* instruments could meet these requirements.

This is why every OCM Exhaust now carries a Marion-made, thermocouple-operated pyrometer on the instrument panel. It indicates the temperature of the exhaust gases; tells when the unit is operating within the safe range. It also measures the air-fuel ratio while idling and so keeps a constant check on engine efficiency.

When you seek meters for unusual or standard applications, it will pay you to investigate Marion. Point for point, Marion invites comparison. Ask for catalog. There is no obligation. Marion Electrical Instrument Company, 415 Canal Street, Manchester, New Hampshire.

Canadian Representative: Astral Electric Company, 44 Danforth Road, Toronto, Ontario, Canada
Export Division: 458 Broadway, New York 13, U.S.A., Cables MORMANEX

marion meters

LOW COST PROTECTION
WITH
GENERAL
FILTERS

ALL-WOOL CLEANABLE
CARTRIDGE
REMOVED FROM
BOTTOM WITHOUT
DISTURBING LINES

Manufacturers of machines using lubricating or hydraulic oils can assure their customers of longer machine life and lower maintenance costs by specifying General AF-500 Filters. They can be installed with inlet at the top or bottom for better space and piping accommodations without altering flow rate or efficiency. Removes solids as fine as five microns and has a capacity of 15-20 gallons per minute depending upon viscosity.

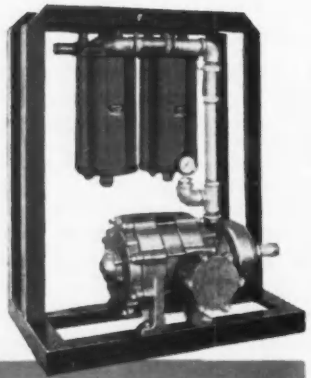
Write for complete specifications and prices. Let us show you why the big industrial users specify GENERAL.

FINEST ALL-WOOL CARTRIDGE

This all-wool felt cartridge is the finest money can buy. Removable from the bottom without disconnecting any lines. Cleanable in minutes in any suitable solvent.

MODEL AF-2000A FOR
TANK APPLICATION

For pumping from a central supply tank or for cleaning vats and tanks the model AF-2000A combines a pump and double filter. Capacity up to 45 gpm depending upon pressure and viscosity. Available with either explosion or drip-proof motors.



GENERAL FILTERS, INC.
12890 WESTWOOD • DETROIT 23, MICH.

machine can understand. The relationships between machine capabilities and the required mathematical tools are illustrated by examples.

Here is the easy-to-understand and practical technical book that gives you a full coverage of the latest information available on the subject—a volume to help scientists, statisticians, and administrative personnel in knowing the capabilities of these devices and in deciding what type of computing system is most readily applicable to a given problem.

Chapters are included on—Counters as Elementary Components; Switches and Gates; A functional Approach to Machine Design; Large-scale Digital Computing Systems; Analog Computing Systems; and Factors Affecting Choice of Equipment.

Structural Plastics

H. C. ENGEL, C. B. HEMMING, H. R. MERRIMAN, 301 *pages, 6x9 in. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York* 18, N. Y. $4.50.

The past ten years have seen a tremendous increase in the number of applications of synthetic plastics. Today there is no industry which is not to some extent dependent upon the utilization of at least several of these relatively new materials. In spite of their widespread use, plastics have not yet achieved, the authors claim, a sound position among engineering materials employed in load-bearing structures. However, with increased knowledge of the limitations as well as the advantages of synthetic engineering materials, increased usage will result whenever cost considerations prove more favorable.

Three factors have been largely responsible for the retarded status of plastics for structural purposes. The first of these are due to certain inherent shortcomings, such as lack of stiffness, toughness, ductility, and dimensional stability under varying environmental conditions. Second, there have been many fabricating difficulties in utilizing this new class of materials. Third, improper use of plastics has been responsible for the failure of many applications. The limitations of such materials must be recognized and taken into account for satisfactory designs.

The lack of reliable information has very seriously hampered the proper structural application of synthetic materials and has been largely responsible for the resistance against structural plastics which is encountered in engineering circles. This book has been

The finish that introduced the furnace into playroom society

Problem: Design a mar-resistant, heat-resistant finish for furnace jackets that would fit in with modern rumpus rooms, make furnaces as colorful and attractive as furniture.

Solution: Industrial Finish Engineers developed special finishes in striking colors. These finishes resist heat, scratching and marring, are easier to keep clean. They go on quickly, dry faster. Their beauty has helped increase sales.

What industrial finish means to sales!

Here is a paragraph from a letter received from one of the largest furnace manufacturers in the country:

"We believe that the finish of our product is of great importance, because that is the part that people see. It plays a great part in selling customers on your product."

We repeat, this is one of the country's largest furnace manufacturers.

A part of their success is due to Industrial Finish Engineering. Find out today how it can increase the saleability, lower production costs of *your* product.

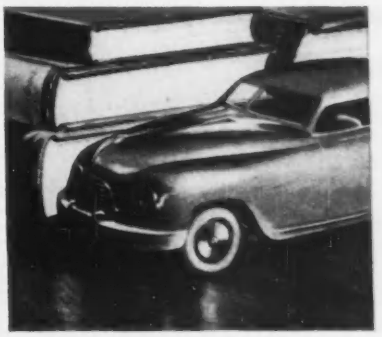
The industry has technical ability to make finishes to suit your specific requirements. Remember, more and more of your customers *start buying with the finish in mind.*

The better the finish – The better the buy!

© 1950, National Paint, Varnish and Lacquer Association, Inc., Washington, D. C.

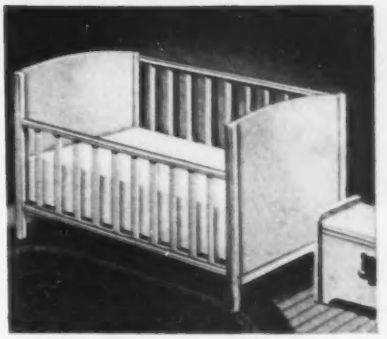
Problem: Find a finish for washing machines that would not be affected by soap and washing powders.

Solution: Industrial Finish Engineers developed special finishes that resist soaps, washing powders and hot water. These quick-drying and baked finishes can be applied after assembly, speed production, cut costs.



Problem: How to equip toy plastic cars with authentic-looking chrome bumpers, without upsetting production.

Solution: Industrial Finish Engineers developed a finish that will not attack the plastic—dries in one minute for continuous production. It improves saleability, keeps costs down.



Problem: How to eliminate the expense of spraying an undercoater over the stain on juvenile furniture.

Solution: Industrial Finish Engineers developed a dip method that used a heavily built-up stain. Result—production savings; a final finish equal in quality to the original method.

EVERY Facility For
PRECISION OPERATIONS
• Required: punch press, multiple drill press, surface grinder, milling machine, turret lathe, drill press, polishing stand, selective plating, rotary grinder, heat treat (selective carburizing, sand blast, harden and draw), hone, cylindrical grinder, hand burring, multiple inspection.
• Required: six-spindle automatic screw machine, bench grinder, centerless grinder, turret lathe, drill press, punch press, internal grinder, heat treat (passivating), multiple inspection.
• Required: six-spindle automatic screw machine, bench grinder, centerless grinder, turret lathe, boring mill, horizontal mill, drill press, super finisher, thread grinder, multiple inspection.

Regardless of how many or what type operations are called for to produce your precision ground parts, Allied has the machinery, equipment—and proven ability—to perform these operations . . . quickly . . . economically . . . and to your most exact specifications.

Send your parts prints please and we will promptly submit quotations.

ALLIED PRODUCTS CORPORATION
DEPARTMENT B-3
12639 BURT ROAD • DETROIT 23, MICHIGAN

PRECISION • EXPERIENCE • DEPENDABILITY
ALLIED PRODUCTS CORPORATION

HARDENED AND PRECISION GROUND PARTS • ALLITE ZINC ALLOY DIES • R-B INTERCHANGEABLE PUNCHES AND DIES • STANDARD CAP SCREWS • SPECIAL COLD FORGED PARTS • SHEET METAL DIES FROM THE LARGEST TO THE SMALLEST • JIGS • FIXTURES

written to provide theoretical and practical information concerning the nature, properties and utilization of the structural plastics. Much of the material given has never been published.

Beginning with the theoretical aspects of high polymeric substances and resins utilized in structural plastics, chapters are given on reinforcing materials and laminated plastics. Low-pressure molding is described in great detail, and considerable data is given concerning sandwich constructions and honeycomb structures. Design equations are stipulated to determine the rigidity and strength of such structures. Considerable test data are used to supplement the analytical treatment; many curves and figures make this material more useful and easily remembered. There is a chapter on structural adhesives which lists not only the composition of the many adhesives available, but the procedure to follow in their application. Included is a chapter on the complex subject of radome design and fabrication.

The appendix describes the various test methods which are used to determine the strength of sandwich materials. Following each chapter is an extensive bibliography which should be quite useful. In addition to the subject index, there is also an author's index.

Falk's Graphical Solutions

KARL H. FALK, 9 x 6 *in.*, 419 *pages; third edition. Published by Columbia Graphs, Columbia, Connecticut.* $6.

This book is a collection of more than 375 worked-out problems that have been solved by the graphical method. Each page represents an infinite number of solutions to one or more particular types of problems.

The examples included have been chosen with the purpose of giving the book the widest possible range of practical application. Examples from mechanics, hydraulics, electricity, construction, physics, chemistry, mensuration, measures and weights have all been included with the aim of giving the user a handy reference book containing solved examples of the types of problems he is most likely to meet in his daily work.

All graphs have been constructed so that only straight-line connections are necessary. Each graph has been worked out to its most simplified form so that the diagram can be read accurately. Each page gives an example of the use of the graph involved, and dashed lines on the graph illustrate sample values. Any combination of factors can thus be solved.